

Proceedings of:

***The Twenty-fourth Annual Meeting of
The American Society for Precision Engineering***

***October 4 to 9, 2009
Monterey Marriott Hotel
and Monterey Conference Center
Monterey, California***

The American Society for Precision Engineering (ASPE) is a multidisciplinary professional and technical society concerned with research and development, design, manufacture and measurement of high accuracy components and systems. ASPE activities encompass relevant aspects of mechanical, electronic, optical and production engineering, physics, chemistry, and computer and materials science. Membership is open to anyone interested in any aspect of precision engineering.

Founded in 1986, ASPE provides a new focus for a diverse but important community. Other professional organizations have covered aspects of precision engineering, always as a sideline to their principal goals. ASPE is based on the core of generic concepts necessary to achieve precision in any application; independent of discipline, ASPE intends to be the focus for precision technology — and to represent all facets from research to application.

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Preface

This book comprises the proceedings of the 24th ASPE Annual Meeting. The contributions reflect the authors' opinions and are published as presented to ASPE, without change. Their inclusion in this publication does not necessarily constitute endorsement by the ASPE, or its editorial staff.

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Welcome Note

Welcome to Monterey and the Twenty-fourth Annual Meeting of The American Society for Precision Engineering. The growth and success of the Society reflects the increasing importance of precision engineering in a wide variety of fields, from manufacturing to microelectronics to basic science. The Society serves as a focus for precision engineers across all of these fields. The Annual Meeting has evolved into a premier international forum for the exchange of ideas and presentation of research results relating to precision engineering, metrology, controls, and system integration. Precision engineers and scientists from private industry, government laboratories, and universities meet to learn about the latest developments and to exchange ideas about the future directions of these technologies.

The 2009 ASPE Annual Meeting continues the tradition of offering two full days of tutorials presented by some of the foremost precision engineers and scientists in the world. The technical sessions and poster presentations offer the latest research results in the areas of ultraprecision machining, metrology, controls, precision grinding, micro-positioning, material processing, design, precision transducers, surface profilometry, and error compensation. The commercial exhibits will provide attendees with the opportunity to view and discuss the latest precision engineering equipment, products, and services.

The conference organizing committee is proud to present the program for the Twenty-fourth Annual Meeting of the ASPE. We welcome your participation and your feedback on how to make future meetings even better.

2009 Organizing & Technical Program Committee

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2009 Annual Meeting

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Program

2009 ASPE Annual Meeting

<i>Time</i>	<i>Sun., Oct. 4</i>	<i>Mon., Oct. 5</i>	<i>Tues., Oct. 6</i>	<i>Wed., Oct. 7</i>	<i>Thurs., Oct. 8</i>	<i>Fri., Oct. 9</i>
8:00	Tutorials	Tutorials				Technical Tours
9:00			Technical Session I	Technical Session III	Technical Session VII	
10:00			Break	Break	Break	
11:00			Technical Session II	Technical Session IV	Technical Session VIII	
12:00	Tutorials	Tutorials	Lunch and Committee Meetings	Awards Lunch and Business Meeting	Lunch and Open Forum	
1:00						
2:00			Commercial Session	Technical Session V	Technical Session IX	
3:00			Break	Break		
4:00			Poster Session I	Poster Session II		
5:00				Hospitality Hr.		
6:00		Welcome Reception	Hospitality Hr.	Technical Session VI		
7:00		Keynote Address				
8:00			Dinner at Monterey Aquarium			
9:00						

☐ Exhibit Open Hours

Keynote Address

Edward I. Moses, Ph.D

*National Ignition Facility and Photon Science
Lawrence Livermore National Laboratory*

Monday, October 5, 7:00 p.m.

Session Chair: Pradeep K. Subrahmanyam, RAPT Industries, Inc.



Dr. Ed Moses is the Director for the National Ignition Facility and the Principal Associate Director at Lawrence Livermore National Laboratory (LLNL) in Livermore, California responsible for photon science and applications. Dr. Moses was responsible for completing construction and activation of the National Ignition Facility (NIF), the world's largest and most energetic laser system and transforming into an experimental platform for the broad national and international scientific user community. Experiments on NIF will access high energy density and fusion regimes with direct application to strategic security as well as applications for fusion energy research, high energy density science, and astrophysics. Dr. Moses is the National Director of the National Ignition Campaign to achieve fusion ignition in the laboratory, the culmination of a 50-year quest. The NIF and Photon Science principal directorate is also responsible for the development of advanced diagnostics and laser technologies for homeland security, competitiveness, and energy needs.

Dr. Moses is internationally recognized in laser and optical sciences. He received a B.S. in 1972 and Ph.D. in 1977, both in Electrical Engineering, from Cornell University. He holds several patents in laser technology, fusion and fission energy, and computational physics. He has received many honors, including the Fusion Power Associates 2008 Leadership Award, the National Nuclear Security Administration Defense Programs Award of Excellence, the Memorial D.S. Rozhdestvensky Medal for Outstanding Contributions to Lasers and Optical Sciences, and the R&D100 Award for the Peregrine radiation therapy program. Dr. Moses is a member of the National Academy of Engineering and a Fellow of SPIE and the American Association for the Advancement of Science.

LIFE: A Path for Sustainable, Carbon-Free Energy Future

The National Ignition Facility (NIF), the world's largest and most powerful laser system for inertial confinement fusion (ICF) and for studying high energy density science, is now operational at Lawrence Livermore National Laboratory. NIF will reach conditions never before achieved in the laboratory (temperatures of ~100 million K, radiation temperatures >3.5 million K, and densities of 1,000 g/cm³.) In March 2009, NIF achieved a 1.1-MJ shot. This achievement made it clear that NIF is not only operational, but is an ignition-capable facility.

Successful demonstration of ignition and net energy gain on NIF will be a major step towards determining the feasibility of ICF and will focus the world's attention on the possibility of laser fusion energy options. Achieving ignition will be enabling for Laser Inertial Fusion Energy (LIFE), a promising laser fusion-driven energy option that we have been exploring.

This talk will focus on NIF's scientific potential and introduces an LLNL-developed approach to generating carbon-free, safe, sustainable and economically competitive power that builds on NIF technologies. The LIFE engine provides a neutron-rich fusion source and can be integrated into a system capable of generating several thousand megawatts of power that is passively safe and minimizes proliferation concerns associated with the nuclear fuel cycle. As such, LIFE can provide a sustainable, carbon-free energy generation solution in the 21st century.

This work performed under the auspices of the U.S. Department of Energy by Lawrence Livermore National Laboratory under Contract DE-AC52-07NA27344.

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<i>Machine Dynamics Research Lab. – Penn State University</i>	

Commercial Session

Tuesday, October 6, 1:30 - 3:00 p.m.

*Session Co-Chairs: Henny A. M. Spaan (IBS Precision Engineering BV)
and Mark T. Kosmowski (Electro Scientific Industries, Inc. - ESI)*

The Commercial Session will take place early in the week on Tuesday afternoon. This is a special session where companies and participants are free to open discussions on a less-formal basis and to promote interaction on a variety of topics. In the first part of this session, company representatives are invited to make brief presentations (five minutes) on new products associated with precision engineering. This provides participants with the opportunity to receive timely information on new technologies that have been commercialized into products and services, as well as information on advances that can be expected in the near future.

2009 ASPE Lifetime Achievement Award



Robert E. Parks

Mr. Parks received a BA and MA in physics from Ohio Wesleyan University and Williams College, respectively. He started his career in 1967 at the Hawkeye Plant at Eastman Kodak Company as an optical engineer and after two years went to work at Itek Corp. where he was an optical test engineer. One of his jobs was to align and test the 6 matched cameras for the S-190 Space Lab project.

From there he went to work for Frank Cooke, Inc. for four years where he got his hands dirty learning how to make optics. This gave him the background to become manager of the optics shop at the College of Optical Sciences at the Univ. of Arizona in 1976. He was at Optical Sciences for 12 years and it gave him the opportunity to write about the projects in the shop and the optical fabrication and testing techniques used there including absolute testing and the installation of a 5 m swing precision optical generator.

Mr. Parks left the University in 1989 to start a consulting business specializing in optical fabrication and testing. Among his jobs was working for the Allen Board of Investigation for the Hubble Telescope and he stayed in residence at HDOS for the duration of the investigation. In 1992 he formed Optical Perspectives Group, LLC as a partnership with Bill Kuhn, then a PhD student at Optical Sciences.

While still at Optical Sciences, Bob became involved in standards work and for twenty years was one of the US representatives to the ISO Technical Committee 172 on Optics and Optical Instruments. For two years he was the Chairman of the ISO Subcommittee 1 for Fundamental Optical Standards. For the last two years Bob is back part time at Optical Sciences helping support optics projects and teaching as part of the Opto-Mechanics program.

Bob is a member of the Optical Society of America, a Fellow and past Board member of SPIE and a member and past President of ASPE. He is author or co-author of over 100 papers and articles about optical fabrication and testing, and co-inventor on 6 US patents.

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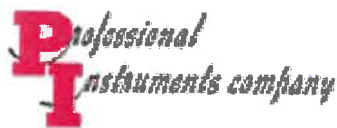
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Technical and Poster Sessions Index

The technical program of the 2009 ASPE Annual Meeting contain 145 papers on precision engineering advances. Papers that lent themselves to significant verbal interaction, concise but important discoveries, or strongly visual or tactile subjects have been selected for the poster presentation. Authors will be in attendance to discuss their work during Poster Sessions I and II on Tuesday, October 6, and Wednesday, October 7 respectively, from 3:30 to 5:00 p.m.

Session I

Commodization of Precision - Challenges and Opportunities

Tuesday, October 6, 2009, 8:30 am - 10:00 am

Session Chair: Pradeep K. Subrahmanyam (RAPT Industries, Inc.) and Thomas A. Dow (North Carolina State University)

1. **Advances in Areal Density for Disk Drives (Invited Paper) ****
New, R. (Hitachi Global Storage Technologies)
2. **A Short History of Diamond Machining (Invited Paper) ***
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3. **From Semiconductors to Photovoltaics (Invited Paper) ****
4. **Nanotechnology (Invited Paper) ****
Teague, E. C. (National Nanotechnology Coordination Office)

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Design of Precision Machines and Instruments I

Tuesday, October 6, 2009, 10:30 am - 12:00 pm

Session Chair: Mark A. Stocker (Cranfield Precision) and Eric S. Buice (Delft University of Technology)

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2. **Design of a Six Degree of Freedom Nanopositioner for Use in Massively Parallel Probe-based Nanomanufacturing**
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3. **Quick-repeat Coupling: A Repeatable Work-holding Fixture for Ultra-precision Turning and Grinding Machines**
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Brouwer, D. M.; Meijaard, J. P.; Jonker, J. B. (University of Twente) 30

**No Abstract available

*Extended Abstract unavailable at press time

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Session Chair: Eric R. Marsh (The Pennsylvania State University) and Christopher S. Margeson
(Nikon Research Corporation of America)

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Session Chair: Vivek G. Badami (Zygo Corporation) and Jonathan D. Ellis (Delft University of Technology)

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Session Chair: Byron R. Knapp (Professional Instruments Company, Inc.) and Robert D. Grejda (Corning Tropol Corporation)

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Design of Precision Machines and Instruments II

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Session Chair: Keith Carlisle (Lawrence Livermore National Laboratory) and Deming Shu (Argonne National Laboratory)

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Session Chair: David D. Gill (Sandia National Laboratories) and Luis A. Aguirre (3M Company)

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Technical Papers



A S P E

A SHORT HISTORY OF DIAMOND MACHINING

**Marsilius, Newman M.
Moore Tool Company, Inc.
Bridgeport, CT 06607**

The technology once generically known as Single Point Diamond Turning has evolved to the point where Ultra-Precision Machining is a probably more appropriate title. With its origins limited to key Government Laboratories and a handful of commercial applications, the core technology has advanced over the past 20 years, significantly in the last decade.

With roots in the initial commercial applications of the 1970's such as photo-copier drums and 14 inch hard drives, everyday technology such as CD/DVD and camera optics, combined with ground breaking advances in illumination optics, imaging systems and medical devices, amongst a few, make Ultra-Precision Machining a key enabling technology for the future.

The paper will provide an insight and will focus on the origins of the technology originally

developed to support applications those various Government Laboratories, also the subsequent proliferation of commercial applications, citing new consumer and industrial products that have been made possible through Ultra-Precision Machining technology; some more obvious than others. It will attempt to show the improvement in both form accuracy and finish capability as contrasted with the reduction in price over the years since the early stages of diamond machining.

If arguments can be developed and articulated, the paper will contrast the evolution of conventional machining with that of Ultra-Precision Machining over the same time period, as well as the contribution of Precision Engineers and Universities to the field as a whole.

DEVELOPMENT OF A NOVEL, ULTRA-PRECISION SPINDLE

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INTRODUCTION

Over the years, several types of spindles have been designed to achieve a precision axis of rotation. Depending on how the spindle couples to its housing, these designs are typically categorized as either contact or non-contact based spindle bearing designs. For example, a journal bearing employs two races centered with respect to the axis of rotation. Typically journals are made from self lubricating alloys [1], and, at high speeds, are designed to take advantage of a hydrodynamic lubrication phenomenon [2,3]. A second example comprises rolling element bearings, to reduce friction at low rotation speeds. The accuracy of those bearings is dependent on precision manufacture of rolling elements and races, preloading of bearings, often resulting in spindle errors ranging from around 0.1 to a few micrometers [4,5], and is a popular solution for precision spindle designs at these performance levels [6-8]. However, more demanding applications such as precision roundness measuring machines require the spindle accuracy to be in a range of 1-100 nm, [9,10]. Non-contact based bearings such as magnetic [11,12], aerostatic [13], hydrostatic [14,15] or hybrid designs [16] are employed, although at the expense of complex designs requiring elaborate controls and/or bearing medium supply.

In recent decades, air bearings have been used extensively for precision applications and represent the most accurate spindle currently available. They are used in variety of applications such as machine tool spindles and are often employed within metrology equipment [9,10]. However, precise manufacturing tolerances as well as the need for a supply of pressurized fluid makes this solution expensive. Moreover, the air supply might introduce a temperature gradient to the metrology frame that has a direct effect on the overall uncertainty of the system. On the other hand, instrumentation spindle requirements for many metrology solutions do not require high rotation speeds or

loads. Additionally, metrology applications can tolerate a repeatable synchronous error motion since it can be mapped and removed during spindle operation. The asynchronous error motions, related to repeatability and accuracy of spindle rotation, are limiting factors and significantly contribute to the overall accuracy of precision spindles.

Alternative instrumentation spindle designs have recently been developed to employ contact pads to form kinematic constraints on a precision shaft [17]. The aim of this paper is to discuss the development of an axially symmetric spindle design employing contact based bearings thereby eliminating the need for an air supply. Characterization studies presented in this paper show that the repeatability of the asynchronous error motion of the newly constructed spindle is in the region of 2 nm which, for metrological applications can provide comparable performance to an air bearing based spindle.

PRINCIPLE OF OPERATION

The presented spindle employs rubbing contact pads symmetrically arranged about the spindle axis, Figure 1. In general, it comprises a solid shaft that is attached to two hemispheres used as a reference for spindle rotation. As mentioned above, the shaft assembly is contacted with the housing through six contacts arranged symmetrically about the axis of revolution of the spindle with all contacts being at the same radial distance from the axis of the spindle rotation. Theoretically, to provide a single freedom of motion only five of the six freedoms of a solid body need to be rigidly constrained. This is achieved by providing five appropriately located rigid contacts. Three contacts are located on one end of the shaft (the reference sphere), and two are located on the other end. An additional sixth contact is provided for the purpose of both applying the necessary force to ensure that the five contacts do not separate during rotation and also to produce a contact force and frictional

forces that are nominally similar to the other contacts. In this manner, a vector sum of all forces between the spindle and housing produce a torsion moment having an axis co-linear with the desired axis of the spindle.

The contact pads are made from the PTFE/Pb based polymer keyed into a porous bronze substrate, and placed at an angle with respect to the axis of rotation. The PTFE material has previously been used for linear translation stages. Based on research on polymer-based bearings it is known that these are capable of producing smooth and repeatable motions at sub-nanometer levels, see Smith et al. [18-21].

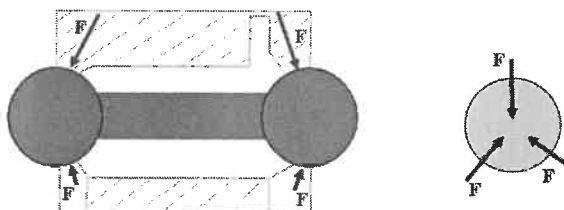


FIGURE 1. Model of the spindle design, where F indicates contact points between pads and the spindle housing, a) side view of the spindle diagram b) is a front view of symmetrically placed contacts.

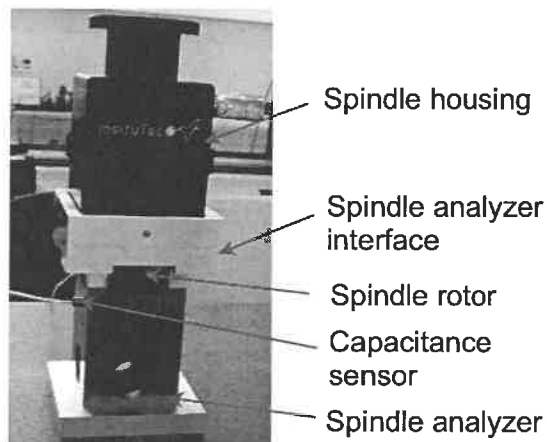


FIGURE 2. Image of the measurement setup.

SPINDLE CHARACTERISATION TECHNIQUE

To assess the spindle error motion, various techniques can be applied. The most popular are; multipoint methods [22,23], Estler reversal [24], and Donaldson reversal [25]. All methods require a reference artifact to measure spindle's motion and therefore are limited to the accuracy of the artifact itself, metrology setup, and equality of sensors sensitivity. As a consequence, it is difficult to properly characterize the synchronous spindle error

motion smaller than around 10 nm, especially to separate spindle error motion from error of the artifact's geometry itself, for example reversal methods require precision rotation of artifact and sensor location by 180° which is difficult to achieve with nanometer uncertainty. Because, the performance of instrumentation spindles are dominated by size of the asynchronous error motion results of repeatability of spindle rotation using custom spindle analyzer equipped with Lion Precision™ capacitance sensors are presented.

MEASUREMENT SETUP

The measurement setup comprises the spindle portion of InsituTec's AccuSurf 3D gauge head. The spindle is equipped with frameless motor from Aerotech, custom low noise slip-ring from InsituTec, Heidenhain encoder and signal conditioning interfacing electronics. The motor is operating in closed loop with an encoder signal. The spindle is capable of continuous rotation and transfer of multiple signals from the stationary housing to the head of the rotating spindle, allowing attachment of custom instrumentation to the rotating shaft. As described above, the rotor is constrained on 6 polymer bearing pads placed symmetrically around the axis of rotation 3 on one end and 3 on the other. The spindle was rigidly mounted to a spindle analyzer stand using a machined interface, Figure 2. On the free end of the spindle rotor a precision reference sphere having a 12.7 mm diameter (supplied by Bal-Tec) and $0.3 \mu\text{m}$ out of roundness error is used. The radial error motion is measured using Lion Precision capacitance gauges with 0.7 nm resolution which are mounted into the spindle analyzer frame.

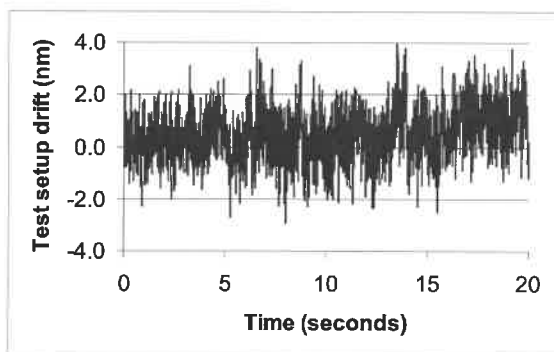


FIGURE 3. Measurement setup noise.

Raw data is presented over a 20 second period showing the sensor's drift less than 6 nm

peak to valley while targeting a fixed position on the artifact sphere, Figure 3. The measurement was performed at NIST in 0.01°C class lab. The assembly was placed on a granite table located at the dimensional metrology laboratory.

SPINDLE MEASUREMENTS RESULTS

Due to space constraints only a small sample representing typical measurement data will be presented. The data collection was performed using a 16 bit DAQ card collecting data at a sampling rate of 10 ms. Therefore to ensure synchronization in the angle between consecutive rotations, Matlab code was written which interpolated collected data at equispaced angular steps. It is important to notice that, at certain spots, the measurement was affected by local spikes originating from scratches, as well as possible particles on the reference sphere. The measurement of the spindle shows 5 μm synchronous error motion coupled with artifact form error, Figure 4. The zoom in on 29 consecutive traces shows less than 10 nm peak to valley data spread with some localized data spikes assumed to be due to regions of higher roughness on the sphere, Figure 5.

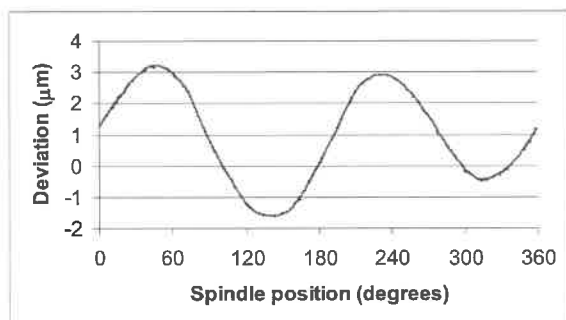


FIGURE 4. Measured error motion combined with artifact form error. Shown are 29 unfiltered consecutive traces measured at 3.7 RPM.

The RMS calculated at each angular position from 29 consecutive traces of data collected while the spindle was spinning at 3.7 RPM and data was filtered using a 50 UPR spline filter. These tests show a 2-3.5 nm asynchronous motion deviations with occasional spikes ranging up to 6 nm, Figure 6. Next, the spindle speed is reduced to 1.5 RPM and similar studies repeated, Figure 7. The slower speeds enable denser data to be collected throughout and as a result the RMS is reduced to 1-2 nm with local spikes observed up to 5 nm.

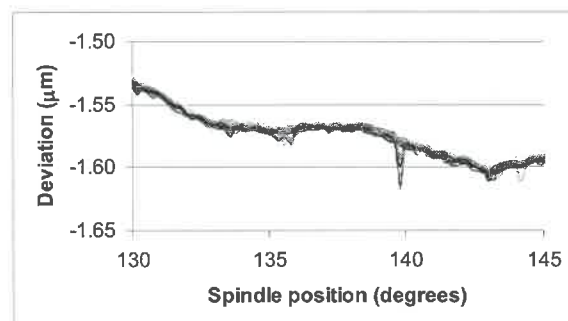


FIGURE 5. Zoom in on portion of 29 consecutive rotations – unfiltered data (~ 8 minutes time).

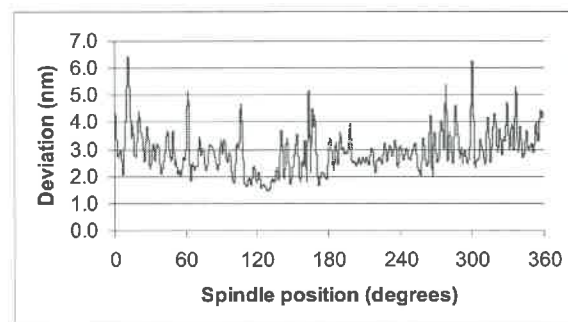


FIGURE 6. RMS of 29 consecutive traces at 3.7 RPM filtered with 50 UPR spline filter. Data taken at constant time intervals producing 4-5 data points per degree.

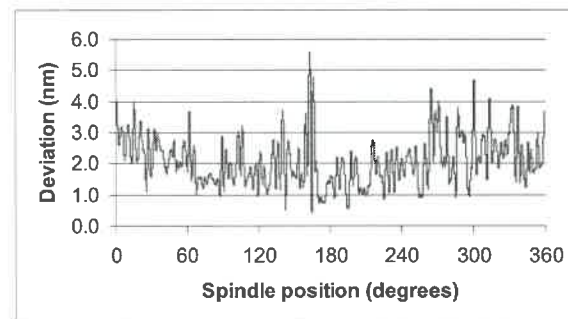


FIGURE 7. RMS of 8 consecutive traces at 1.5 RPM filtered with 50 UPR spline filter. Data taken at constant time intervals producing 11-12 data points per degree.

Additionally, a long term test was performed for a duration of 15 hours, Figure 8. The spindle was rotating at approximately 3.7 RPM and 3300 revolutions were recorded. The calculated RMS showed 10-15 nm asynchronous error motion which largely could be due to mechanical as well as an electrical drift of the components over such long period of time.

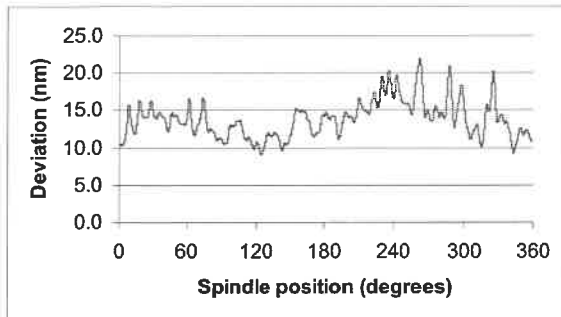


FIGURE 8. RMS of 3300 consecutive traces at 3.7 RPM filtered with 50 UPR spline filter. Data taken over 15 hours.

ACKNOWLEDGMENTS

Authors would like to thank Dr. Bala Muralikrishnan from NIST for great help during the final testing phase.

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DESIGN OF A SIX DEGREE OF FREEDOM NANOPositionER FOR USE IN MASSIVELY PARALLEL PROBE-BASED NANOMANUFACTURING

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SYNOPSIS

In probe-based nanomanufacturing (PBM) micro-scale probe tips are used to make, manipulate, or measure nm-scale features on a substrate. Practical throughput rates require 2D tip arrays (up to 55,000 tips) wherein the probe tips lie on a 'surface'. These 'surface' tools require six degrees-of-freedom (DOF) positioning to control relative position and orientation between the tip arrays and substrate.

It may become necessary to run several nanopositioners in parallel to achieve high throughput rates, therefore nanopositioner cost and footprint must be minimized. Small-scale nanopositioners satisfy cost/size requirements, but equipping them with multi-axis sensing that enables practical position control is challenging. Herein we present a meso-scale nanopositioner with integrated piezoresistive displacement sensors that operate as the device moves through its $50 \times 50 \times 50 \mu\text{m}^3$ work volume.

INTRODUCTION

Meso-scale, 6-axis nanopositioners are lower-cost (\$100s vs. \$10,000s US), exhibit smaller thermally-induced errors, and possess higher natural frequencies than their macro-scale counterparts. Their small size enables several nanopositioners to be run in parallel within a small footprint. Existing macro-scale sensors are too large and expensive to integrate within these devices. New sensors are required to enable 6 axis sensing and thereby make possible the manipulation of 2D probe arrays that will yield high throughput in processes such as dip-pen nanolithography (DPN), nano-electrical discharge machining (nEDM), biological assays and other PBM processes.

THE NEED FOR 6-AXIS POSITIONING

Linear arrays of ~ 10 tips have been used to improve throughput, however 10,000X larger throughput will be required for practical PBM [1]. The emerging PBM industry has moved in this

direction, as evidence by NanoInk's 1cm^2 , 55,000 tip arrays [2]. Six-axis positioners enable leveling that prevents non-parallelism errors, depicted in Fig. 1, thereby reducing variations in gap and/or contact forces across the array.

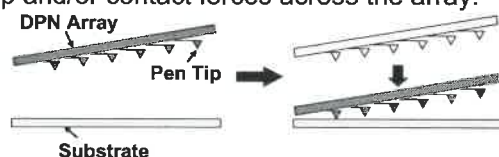


FIGURE 1: Consequence of parallelism error

NANOPositionER DESIGN

The nanopositioner design is based upon the 6 DOF, monolithic HexFlex flexure system [3,4]. Figure 2 shows our meso-scale HexFlex, less actuators and electronics. This design is attractive as (i) its 2½D geometry may be microfabricated and (ii) its in-plane symmetry yields low thermal sensitivity [3].



FIGURE 2: Meso-scale HexFlex nanopositioner

Figure 3 shows how in-plane stage motion is achieved by applying combinations of in-plane forces; and how out-of-plane motions are achieved by out-of-plane forces.

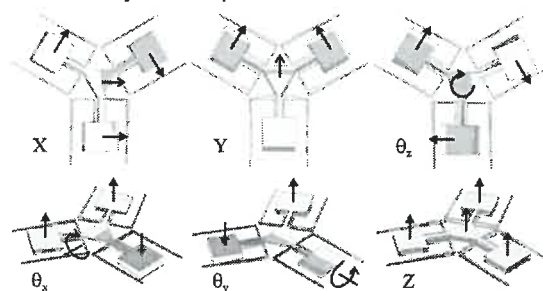


FIGURE 3: HexFlex actuation-motion examples

PROBE-BASED NANOMACHINING CENTER

System architecture

The meso-scale HexFlex was integrated into the probe-based nanomachining center (PNC) shown in Fig. 4. The system was designed to process arrays of DNA [1] with tips. A suitable processing rate requires simultaneous operation of 550,000 tips. The PNC uses 10 nanopositioners that each are equipped with a 55,000 tip Nanolink array. A kinematic alignment interface is used to detachably align each meso-scale HexFlex into a dock as shown in Fig. 4.

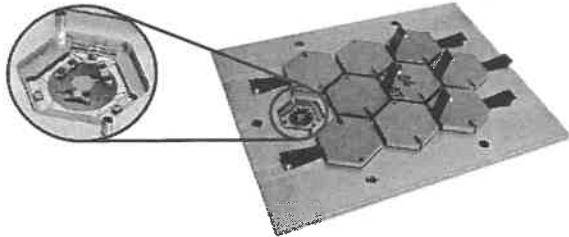


FIGURE 4: PNC with detail of docking station

Actuation

Each dock is covered by a flexible circuit which is attached to the bottom of a hexagonal aluminum heat spreader plate that helps to dissipate up to 18W per HexFlex from the HexFlex's actuators. A flexible circuit (with heat spreader plate removed) is shown in Fig. 5.

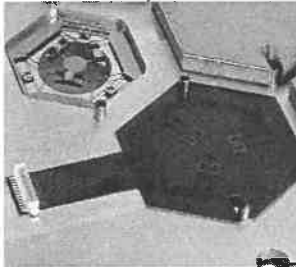


FIGURE 5: Flex circuit and heat spreader plate

Three sets of magnets (with three magnets per set) are affixed to the stage's paddles as shown in Figs. 5 and 6. The magnetic field of each set interacts with a pair of corresponding coils within the flex circuit to impart six independent actuation forces [5-7] upon the HexFlex. The 1st coil in each set imparts in-plane forces and the 2nd imparts out-of-plane force. Coil-magnet spacing was set at 150 μm . Figure 6 shows an image wherein the coils are shown relative to their corresponding magnet sets. Each actuator provides ~ 10 mN of force, thereby enabling a work volume $50 \times 50 \times 50 \mu\text{m}^3$. Control is facilitated by the linearity of the actuators.

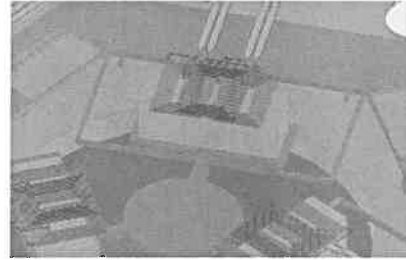


FIGURE 6: Coil-magnets within Lorentz actuator

The linearity of the Z axis response, shown in Fig. 7, is similar to the linearity in other axes.

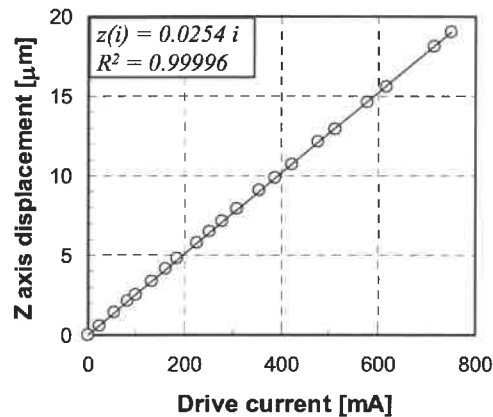


FIGURE 7: Z motion vs. actuator drive current

Sensor selection and layout

Piezoresistive strain sensors [8-13] were microfabricated atop the flexure elements as shown in Fig. 8. The combination of flexures and piezoresistors were used due to (1) ease of integration, (2) microfabrication compatibility, (3) insensitivity to electromagnetic noise/fields, and (4) ability to sense sub-nm displacements.

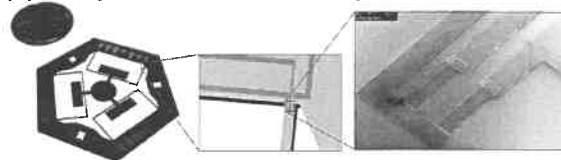


FIGURE 8: Location and shape of sensor

Micro-scale piezoresistive sensors have focused on sensing single direction motions, e.g. bending of a cantilever [16] or flexing of diaphragms [13,14]. A few attempts have been made to measure two-axis displacements [12]; however 6 axis sensing has not yet been demonstrated. In our design, two sensors were placed at the ground of each flexure where the highest strain occurs. This enables sensing with fine resolution and high signal-to-noise (SNR) ratio.

As shown in Fig. 8, the sensors work in pairs. A 1st sensor is placed along the neutral axis of the beam and a 2nd sensor is placed at the beam's edge. The sensor on the beam's neutral axis is subjected to strain from out-of-plane bending while the sensor on the edge of the beam is subjected to strain from in- and out-of-plane bending. Their combined readings may be used to extract the in- and out-of-plane bending of the flexures. A minimum of three pairs, e.g. 6 sensors, enable 6 axis displacement sensing.

As the stage moves, the supporting flexures strain and cause a resistance change, ΔR_{sensor} , in the sensor(s) atop the flexure. The change in resistance is:

$$\Delta R_{\text{sensor}} = \pi_l E \epsilon_l R_{\text{sensor}} \quad (1)$$

In Eq. 1 R_{sensor} is the nominal resistance of the sensor, E is the elastic modulus, and ϵ_l is the strain applied along the longitudinal axis of the sensor. The longitudinal piezoresistive coefficient, π_l , for polysilicon is typically $20 \times 10^{-11} \text{ Pa}^{-1}$ [14, 15]. Given each sensors' strain, Eq. 2 may be used to determine stage displacements.

$$\delta = [\Delta x \ \Delta y \ \Delta z \ \Delta \theta_x \ \Delta \theta_y \ \Delta \theta_z]^T = \mathbf{J} \epsilon \quad (2)$$

In Eq. 2, δ is a vector of stage displacements and ϵ is a vector of the longitudinal strains for each sensor. The Jacobian, $\mathbf{J}$, which links them, is a function of flexure geometry.

Sensor noise

The resolvable displacement is limited by (1) Johnson noise and (2) flicker noise. The fluctuation in voltage, $\Delta V_{\text{Johnson}}$, is given by Eq. 3.

$$\Delta V_{\text{Johnson}} = \sqrt{4k_b T R_{\text{sensor}} \Delta f} \quad (3)$$

In Eq. 3 k_b is the Boltzmann constant, T is the conductor temperature, and Δf is the sensor bandwidth in Hz. Our simulations predict Johnson noise equal to 175nV, or SNR = 19 for 10nm in-plane steps and 26 for 5nm out-of-plane steps.

Flicker noise is generated by the capture and release of charge carriers in a semiconductor. The fluctuation in voltage, $\Delta V_{\text{flicker}}$, is given by Eq. 4.

$$\Delta V_{\text{flicker}} = \sqrt{\frac{\alpha V_{\text{sense}}^2}{N} \ln \left(\frac{f_{\text{max}}}{f_{\text{min}}} \right)} \quad (4)$$

In Eq. 4, V_{sense} is the voltage drop across the sensor, α is the Hooge constant (1×10^{-4} to 1×10^{-3} for polysilicon [17]), f_{max} and f_{min} define the bandwidth of the sensor in Hz, and N is the number of charge carriers in the piezoresistor. Our simulation predicted flicker noise equal to

310nV, or a SNR of 6 with respect 10 nm in-plane steps and 8 for 5 nm out-of-plane steps.

MICROFABRICATION

A 304 nm layer of thermal oxide was grown upon a 150mm diameter, 400 μm thick silicon wafer. This provides electrical insulation for the piezoresistor and metal contact layers. A 530nm layer of N-type polysilicon (resistivity = 0.01749 $\Omega\text{-cm}$) was deposited and patterned on the oxide to form piezoresistors. A 500nm layer of aluminum was deposited and patterned to form electrical contacts. Deep reactive ion etching was used to create the stage and flexures. Wafer and die level images of the HexFlex nanopositioners are shown in Figs. 9 and 10.

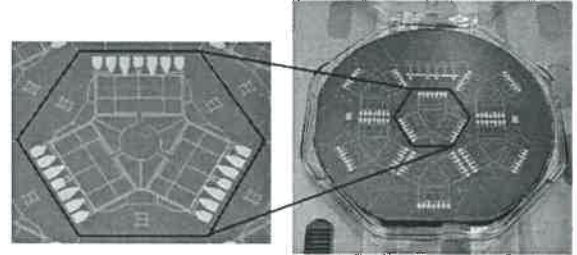


FIGURE 9: Wafer prior to separation of dies



FIGURE 10: mesoHexFlex separated from wafer

EXPERIMENTAL RESULTS

Figure 11 shows the test setup.

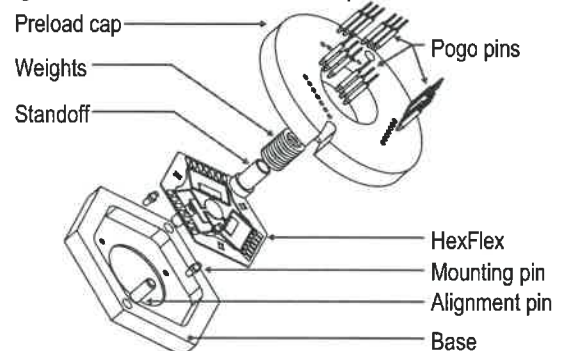


FIGURE 11: Exploded view of testing apparatus

Eighteen pogo pins were used to port electrical signals from the sensors. Weights were stacked upon the device to induce stage deflections. By

changing the orientation of the test setup from horizontal to vertical, out-of-plane and in-plane deflections could be measured. Each piezoresistor was measured via a $\frac{1}{4}$ Wheatstone bridge and the signal was amplified with an AD624 amplifier.

EARLY RESULTS AND DISCUSSION

The results for an in-plane sensor (during an in-plane test) and an out-of-plane sensor (during an out-of-plane test) are shown in Fig. 12.

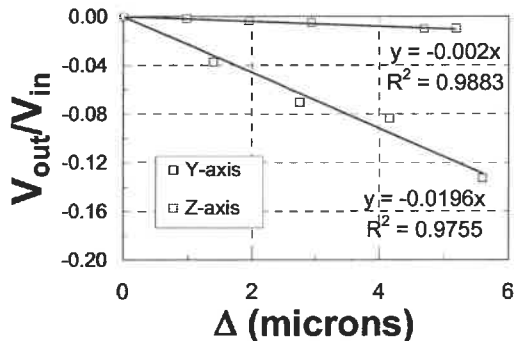


FIGURE 12: Sensor output

In our proof-of-concept experiments, we found that thermal fluctuations affected the stability of the electronics and the sensor's resistivity, thereby causing up to 500nm error in sensor readings. This likely led to the deviation of data points from linear (expected) behavior. The rough adherence to linear behavior indicates that we can use thermal compensation resistors to mitigate thermal errors in a second prototype. Based upon the effectiveness of this approach in other applications, we envision being able to resolve 5 nm out of plane and 10 nm in-plane when sampling at 100 Hz.

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QUICK-REPEAT COUPLING: A REPEATABLE WORK-HOLDING FIXTURE FOR ULTRA-PRECISION TURNING AND GRINDING MACHINES

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INTRODUCTION

A highly repeatable work-holding fixture has been developed to enable fast, in-process inspection between an ultra-precision turning and/or grinding machine and a metrology instrument such as a coordinate measuring machine (CMM). Designed to quickly engage and disengage, the coupling is estimated to repeat its position to $<1\text{ }\mu\text{m}$ while supporting up to a 10 kg payload. The coupling features micro-finished tungsten-carbide spheres and quarter cylinders configured as a three-vee kinematic coupling. The nesting force (or preload) is applied with a quarter-turn latch mechanism acting through a spring plate and cross-axis cam followers to eliminate sliding friction. The coupling also features a labyrinth seal to exclude coolant and debris from contaminating the contacting surfaces. This paper presents aspects of the coupling's design and analysis. Additional information about this product may be obtained by contacting our company [1].

BACKGROUND

The use of kinematic couplings for repeatable fixturing applications was advanced long ago by the renowned scientists James Clerk Maxwell and William Thompson (Lord Kelvin), and is well known today to the precision engineering community. Used typically to establish a repeatable relationship between two bodies, a kinematic coupling provides six independent constraints formed by well-defined contacting surfaces. For example, three spheres on one body resting in three vees on the other body provide three pairs of independent constraints, when properly placed and held in contact with a nesting force. The symmetry of three vees is compelling for many applications, but a better known configuration is formed with three spheres on one body resting in a tetrahedral socket, a vee and a flat on the other body.

Depending on the application, developing one of these two basic configurations into a finished design may range from a catalog order to rather elaborate engineering analysis and design synthesis. Common pitfalls to consider are:

managing high contact stresses, choosing suitable materials, and arranging the six constraints and the nesting force to engage under load with friction. This particular project made use of earlier published work to optimize the coupling configuration to achieve particular performance measures [2,3,4].

THE QUICK-REPEAT COUPLING

Purpose

The *Quick-Repeat Coupling* (QRC) enables the workpiece and its supporting fixture to be moved from the manufacturing machine to the inspection machine and returned for subsequent work without further adjustment or compensation of position or orientation. To be useful, the QRC must be repeatable, robust and rapidly engaged and disengaged.

Description

The Quick-Repeat Coupling typically includes three separate assemblies: the Spindle Mount, the Metrology Mount and the Universal Interface which holds the workpiece and will couple to the two mounts as FIGURES 1, 2 and 3 show. The contacting surfaces, visible in FIGURES 4 and 6, are configured as a three-vee kinematic coupling. For the Spindle Mount, which typically has a horizontal axis, the surfaces are held in contact by a compliant mechanical latch that the operator turns using a T-handle hex key (8 mm or 5/16 in). The simpler Metrology Mount is gravity loaded and can only be used with a vertical axis. In addition, the QRC has the following notable features:

- The QRC is 8 inches [203 mm] in diameter and 2.75 inches [70 mm] in overall length. The masses of the Spindle Mount, Metrology Mount and Universal Interface are: 10.1 lb. [4.6 kg], 2.2 lb. [1 kg] and 13.4 lb. [6.1 kg], respectively.
- The Spindle Mount attaches to the face of a Precitech SP150LC spindle using the same bolts as the standard vacuum chuck.
- The coupling is designed to operate at a maximum spindle speed of 5000 rpm.

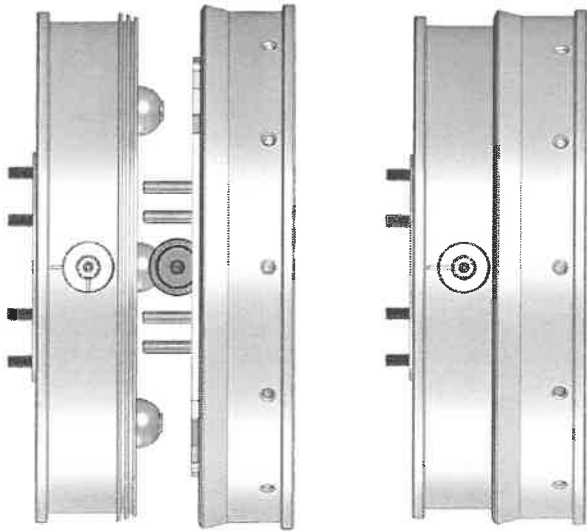


FIGURE 1. Views of the Spindle Mount and Universal Interface, shown unlatched and disengaged (left) and engaged and latched (right), are in the typical orientation for a horizontal spindle, which is not shown but would attach to the left-most face.

- The mechanical latch works as a toggle mechanism with cross-axis cam followers and a spring plate to provide a repeatable nesting force up to 200 lb. [890 N] with minimal friction to influence positioning, see FIGURE 5.
- The mechanical latch has a spring-loaded catch to secure the latched position, which is visibly indicated with alignment marks. To disengage the coupling, the operator must release the catch by pushing in with the T-handle and then rotating $\pm 90^\circ$ to one of the unlatched positions.
- The coupling geometry and materials were chosen for a combination of repeatability, stiffness and robustness. The contacting surfaces are micro-finished tungsten-carbide spheres and quarter cylinders, all with 3/4" inch [19 mm] diameter. The repeatability is estimated to be $<1 \mu\text{m}$ while supporting a 10 kg payload. The stiffness is estimated to be $>200,000 \text{ lb/in}$ [35 N/ μm]. Repeatability, stiffness and load capacity are taken at a point representing the center of the workpiece, 1 inch [25 mm] from the face of the Universal Interface.
- The six contact surfaces critical for repeatability are protected from grinding debris and coolant using a non contacting labyrinth seal, see FIGURE 5. In addition, these surfaces are accessible for occasional cleaning.

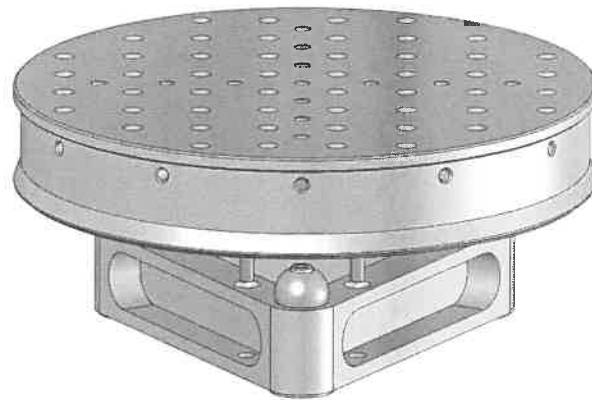


FIGURE 2. The Universal Interface has a square pattern of (52) M5 x 0.8 threaded holes and (13) Ø5-mm pin holes all on 24-mm centers. It may be moved from the Spindle Mount to the simpler Metrology Mount shown here disengaged.

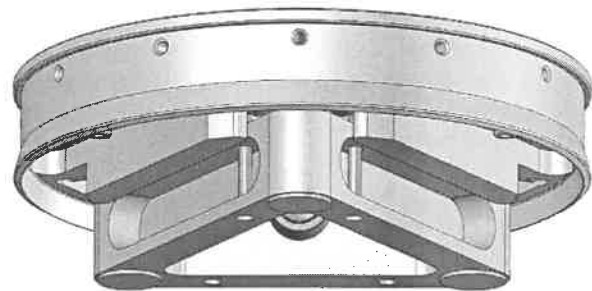


FIGURE 3. The Universal Interface shown engaged to the Metrology Mount is preloaded only by gravity, that is, its own weight plus payload.

- Reference surfaces are provided on both the Spindle Mount and the Universal Interface to: enable concentric setup to the spindle, serve as datums for the CMM and verify repeatability.
- Four asymmetrical guide pins prevent accidentally engaging the coupling in the wrong angular orientation. They also serve as legs if the Universal Interface is placed on a table.
- The Spindle Mount has two adjustable counterweights for balancing as an independent assembly. The Universal Interface has three adjustable counterweights and is balanced while engaged with the Spindle Mount. The workpiece and its supporting fixture are balanced using radial set screws in the Universal Interface (see FIGURES 4 and 6).
- The exterior housings are constructed of hardened stainless steel.



FIGURE 4. Inner workings of the spindle mount include the quarter-turn latch mechanism, three spheres and two adjustable counterweights for single-plane balancing.

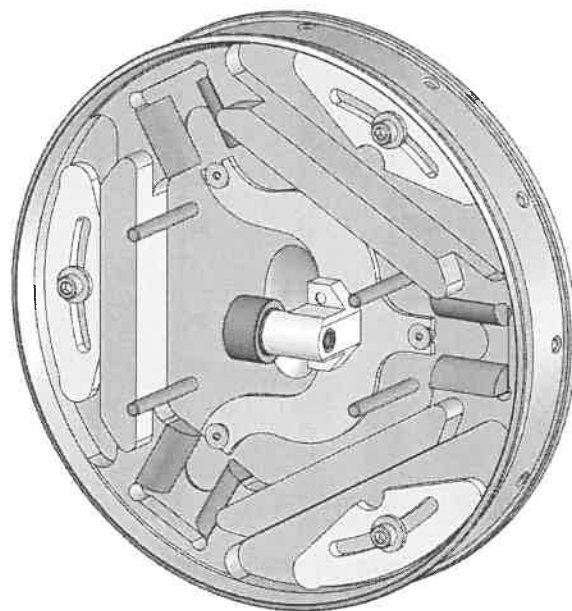


FIGURE 6. Inner workings of the universal interface include a Y-shaped preload spring, three vees formed by six quarter cylinders, and three adjustable counterweights for single-plane balancing.

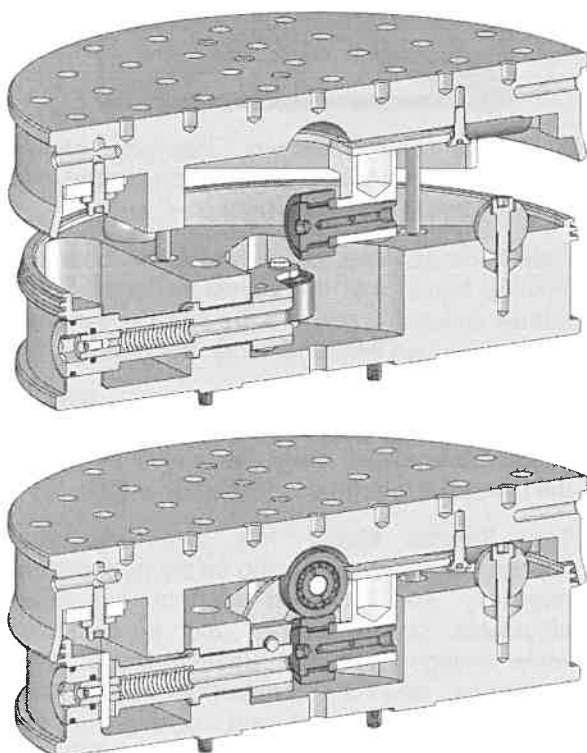


FIGURE 5. Section views show the QRC unlatched and disengaged (top) and engaged and latched (bottom).

Design Analysis

The requirement that the QRC must work on a horizontal-axis spindle makes the design process more challenging for several reasons but in particular, the performance will suffer compared to a more favorable orientation. A horizontal-axis coupling must be preloaded with a higher nesting force to hold all the kinematic surfaces in contact. As a rule of thumb, the ratio of the highest to lowest contact force should be of order 2:1 or less. Higher force leads to proportionally higher friction and greater positioning error because stiffness is increasing only to the one-third power.

The analytical model developed for the QRC estimates the positioning error (or repeatability) based on the real coefficient of friction (assumed to be 0.1) and the stiffness of the contacting surfaces. Using the payload and nesting force stated earlier, the model predicts little change in repeatability between a horizontal axis at 0.875 μm and a vertical axis at 0.818 μm . The significant improvement comes by reducing the nesting force. For example, the repeatability for the gravity-loaded Metrology Mount is predicted to be 0.231 μm .

It is possible that a kinematic coupling can become stuck well out of position when the restoring force is too small to overcome friction in

the contacting surfaces. This is especially true of a coupling with a horizontal axis and is the basis for the rule of thumb stated earlier. An analytical measure of this condition is the limiting coefficient of friction, where the coupling is on the verge of sliding or not. Having a limiting coefficient of friction much larger than the real (or material) coefficient of friction is desirable and can be thought of as a factor of safety.

The analytical model also calculates the limiting coefficient of friction. The QRC oriented with a vertical axis has a 0.36 limiting coefficient of friction independent of the nesting force. Using the payload and nesting force stated earlier (200 lb.), the QRC oriented with a horizontal axis has a 0.25 limiting coefficient of friction. Reducing the nesting force by 2x reduces it to 0.15, which very likely could stick. Increasing the nesting force by an unpractical 10x increases it to 0.35, nearly the value for a vertical axis.

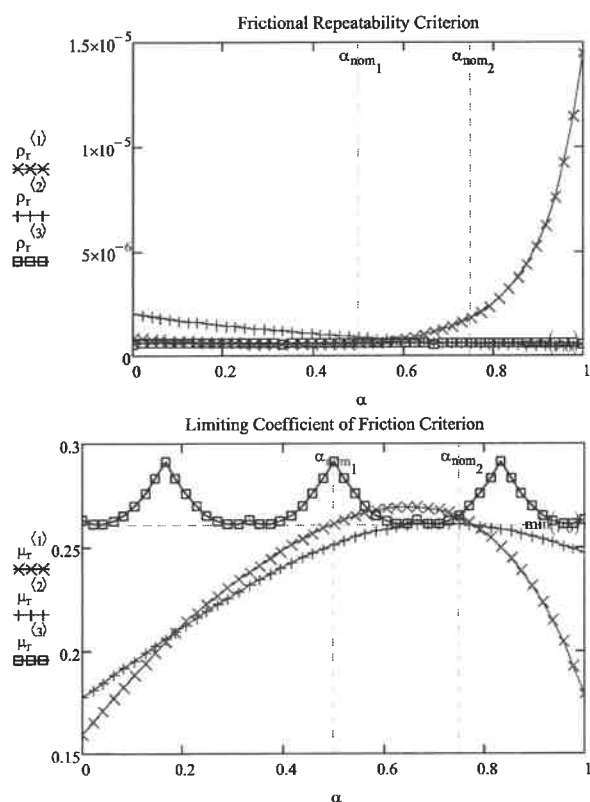


FIGURE 7. Plots of repeatability (top) and limiting coefficient of friction (bottom) vary one parameter at a time from nominal values on a normalized scale. The nominal values are: 60° for inclination angle (parameter 1) and 10° for divergence angle (parameter 2). Parameter 3 is 360° rotation of a horizontal-axis spindle.

These two measures, repeatability and limiting coefficient of friction, may be used to optimize the configuration of the contacting surfaces, the magnitude and/or direction of the nesting force, and other design parameters. The basic configuration of the QRC was selected to be a symmetric three-vee coupling and the size was fairly constrained. Two angle parameters remain to describe the shape of the vee. The obvious one is the inclination angle (the complement of one-half the included angle of the vee). The other angle can be represented a couple of ways but there is a manufacturing preference. The quarter cylinders that form the vee need not have parallel axes; that is, they could have a divergence angle measured from the radial axis of symmetry.

FIGURE 7 shows two plots from the analytical model. Unfortunately space does not permit an adequate explanation. The results indicate there is some benefit to having a divergence angle, especially when the payload is offset further from the coupling face. However for simplicity, the first version of the Quick-Repeat Coupling has parallel quarter cylinders (zero divergence).

ACKNOWLEDGEMENTS

The Quick-Repeat Coupling features kinematic components (counter bored spheres and quarter cylinders) designed and manufactured by Bal-tec. The author is grateful to Eugene Gleason for his advice on this project and for providing a valuable guide to practical kinematic devices [5].

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PROTOTYPE OF A PLANAR GUIDED MILLING MACHINE FOR HIGH PRECISION APPLICATIONS

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ABSTRACT

A planar guided machine for high productivity in milling of high-precision parts is presented. High productivity requires high dynamic feed rates and high stiffness to withstand cutting forces. This is achieved by the introduction of a planar guided stage into the design of standard milling machines. With application of standard machine components like roller bearings the machine combines high productivity with high precision for moderate costs. The concept is explained in detail, a prototype, error-budgeting, simulation results and first measurements are shown.

INTRODUCTION

In the field of machine tool development there is a strong need for improved physical performance, accuracy, stiffness. Beside this, the cost for achieving these improved properties should be considerably reduced compared with the actual values.

Actual machine design for milling machines is dominated by conventional serial arrangements of typically three linear axes. In these concepts it is required to stack at least two axes onto another while one axis has to move at-least also another axis. This brings a limitation in dynamics, stiffness and an addition of errors of at least two guidances. A possible solution is to reduce the number of error-sources by reducing the number of mechanical components in the structure and by this of mechanically restricted degrees of freedom by usage of planar-guided stages [2]. Planar guided X-Y-tables are well known in the IC-industry where no process forces have to be born [3].

The potential of smooth guiding quality with an aerostatic planar guided stage is shown in [4]. The deviations are typically in the sub- μm -area. Investigations in planar multi-coordinate-drives showed a reduction of the deviations of these planar guided stages also in the directions parallel to the guidance plane and possible positioning steps of only a few nm in size [5].

In [6] the application of voice-coil based planar driven stages is shown also for relatively large travel ranges of about 250mm.

These nano-resolution capabilities are possible as a result of planar low-friction drives, however, they are up to now not available for feed-forces of over 100 N. In scaling up the available forces also a cooling problem of the drive-components is shining up [7]. The task is therefore to combine planar guiding with linear drives.

DESIGN

In order to improve accuracy and stiffness and to reduce the cost for milling applications at the same time, a new machine concept is proposed here, called PREZOPLAN. The basis of the PREZOPLAN is a thick granite base plate with a planarity of $0,8\mu\text{m}$ over the whole planar guidance surface. On the top surface of this base plate there is the stage guided on an aerostatic bearing with a gap of about $6\mu\text{m}$ and a total vertical stiffness of $600\text{N}/\mu\text{m}$ at a vertical preload of 3500N . A new manufacturing technology of micro nozzles [8] allows the integration of 4 supporting areas each containing 8 nozzles directly into the bottom side of the stage. These 4 supporting areas are ground with a good planarity. The integration in one block without a special mounting component required increases stiffness and dynamic stability of the aerostatic bearing. The preload is realized with disc springs, but can also be generated by a vacuum chamber arrangement. The stage is moved in the planar axes X, Y by a traverse which is driven by a gantry configuration of ball screw drives in Y-direction. The traverse is guided with linear roller bearing guideways in y-direction mounted onto the granite basis. The stage is assembled to the traverse with elements flexible for an A-, B- and Z- movement and stiff in Y- and C- direction. This flexible element is required because the degrees of freedom in A-, B-, and Z-direction shall be solely determined by the guiding plane. In X-direction there is a ballscrew drive mounted

onto the traverse which moves the stage with a driven spindle nut. This arrangement of drives is optimized for a minimum of thermal load of the machine structure due to the drive motors. The dimensioning of all ballscrew drives is identical for high acceleration of 25m/s^2 minimum, feed rates up to 60m/min and feed forces of 2500N per axis. Due to the alignment of the centers of mass and the drive locations close to the guiding plane, no considerable inertial or frictional cross-talk should appear [9].

Position feedback is effected by linear scales fixed to the granite basis for Y-direction and onto the traverse for X-direction. In figure 1 the current construction of the X-Y-stage is shown. The view is onto the ballscrew drive in X-direction which is mounted onto the traverse and moves the stage by a fixed bearing at the end of the spindle.

To complete the system the stage is integrated into a portal frame consisting of a massive basis to bear the reactional forces of high speed movements, which is put on rubber air mounting elements for low frequency decoupling as seen in figure 2. The portal frame as shown in figure 3 carries a conventional Z – axis. To ensure that the aerostatic guidance will not be contaminated with cutting fluid and chips, there is a planar coverage foreseen.

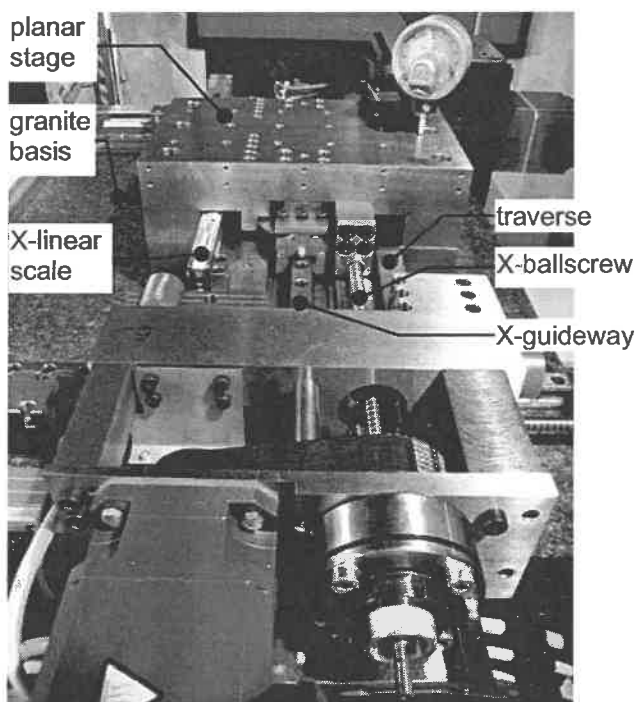


FIGURE 1: planar stage with traverse

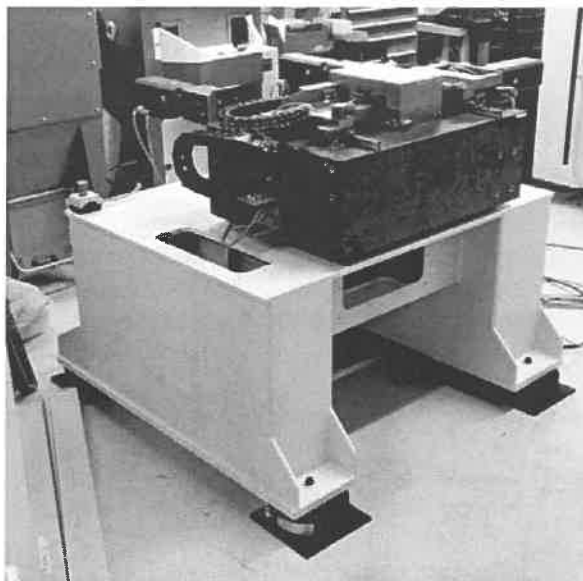


FIGURE 2: two axis machine demonstrator

To set up the workshop capability a housing is actually under construction.

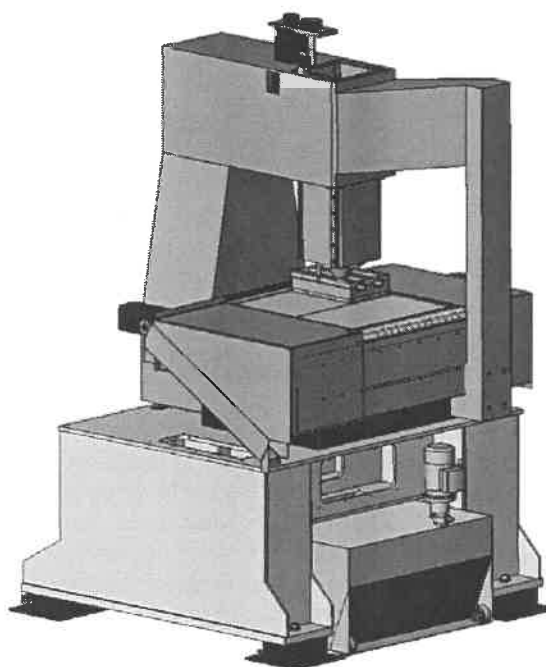


FIGURE 3: three-axis machine PREZOPLAN

ERROR BUDGETING

An Error Budget is used to outline all major sources of error for the machine. The amount of error contributed by each source is quantified and calculated with the relevant geometric values [10, 11]. The PREZOPLAN arrangement is compared with a conventional machine design

containing a stack of two roller guided linear axes with identical working area. These two-axis arrangements are compared to each other as well as the corresponding three-axis arrangements each with the same third axis shown in figure 3.

Basic parameters are

- Encoder span 1,5 μ m (linear scale)
- Straightness of roller guideways 2 μ m
- Straightness of planar guided axes as a result of planarity error of granite 0,8 μ m
- Rotational errors of roller guideways 10 μ m/m
- Rotational errors of planar guided axes 5 μ m/m
- Squareness is set to 0, as it can be compensated because of its systematic character.

The error budgeting results in table 1 and table 2 show that the resulting combined expanded positioning uncertainties can be significantly reduced by planar guiding. Table 1 shows the combined expanded positioning uncertainties to be expected in each direction for the two axes arrangement, table 2 shows the uncertainties for the three axes system.

Two-axis	U_x [μ m]	U_y [μ m]	U_z [μ m]
planar	1.5	1.6	0.6
stacked	2.9	2.8	2.1

TABLE 1 error budget comparison for two axes

Three-axis	U_x [μ m]	U_y [μ m]	U_z [μ m]
planar	3.3	3.3	1.3
stacked	4.1	4.0	2.3

TABLE 2 error budget comparison for three axes

This error budget focuses on the comparison of a planar guided two- and three axis system with a conventional stacked cross table arrangement with identical properties in reachable precision by the same accuracy in manufacturing. For example it is similar to reach a straightness of 2 μ m by roller bearing or 0.8 μ m by aerostatic bearing with granite basis like in the PREZOPLAN-concept. In this case, the results of the error budgeting should be understood as a basis value in comparison to a conventional arrangement. To improve the uncertainties, its possible to reduce the several sources of error. For improved precision as a demand a precision-option is required. It can be reached in using full-aerostatic bearings in all directions for example.

SIMULATION RESULTS

The static and dynamic behavior of the structure including the planar guided stage, base frame and Z-Axis support have been simulated with the Axis Construction Kit (ACK) [12]. In figure 4 the simulation result is shown with the deflection of the stage in X-Axis direction at the first eigenfrequency relevant for positioning at 63Hz. With this appropriate values for dynamics of feed-rate and acceleration with high jerk should be reachable. Furthermore dynamic path deviations will be minimized, caused by cross-talk free alignment of ballscrew drives near the centers of mass.

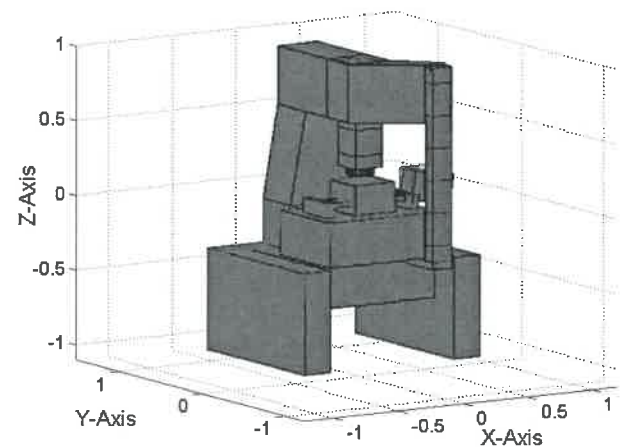


FIGURE 4: First significant mode shape based on ACK-calculations, verified by experimental modal analysis

This and further more basic behavior of the simulation, focused on the base frame with planar stage, have been verified by the results of the modal analysis of the two axis machine demonstrator.

MEASUREMENTS

In Figure 5 and in Figure 6 first measurement results of the straightnesses are shown. The measurements were done with an incremental Heidenhain probe and a straightedge. The short-wave signals are to be led back on the roughness of the granite straightedge while periodicities of 5 and 10 mm are caused by mechanical improperties of the drives.

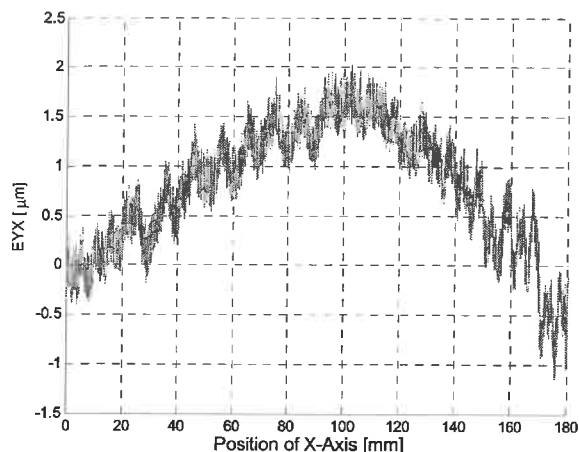


FIGURE 5: Straightness of the conventionally guided X-Axis in the horizontal (Y-) direction

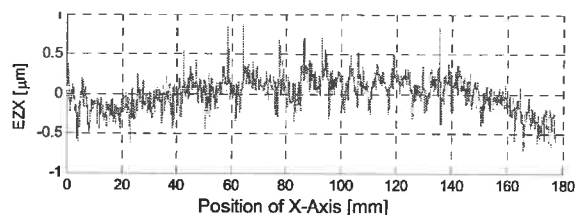


FIGURE 6: Straightness of the aerostatically guided X-Axis in the vertical (Z-) direction

The deviations of the guidance itself are characterized by the flatness of the guidance surface with about $0.8\mu\text{m}$ and/or by the straightness of the conventional guidance with over $1.5\mu\text{m}$. The last one will be improved by using a high-end guide way and with an optimized assembly procedure. The influence of the ballscrew is to be reduced also using a decoupling element between lead screws, traverse and planar stage.

CONCLUSION

The planar guided PREZOPLAN concept for cutting machine tools represents a substantial improvement and offers in relation to the state of the art concepts a reduction of positioning uncertainties and an increase in the stiffness as well as a simplification and reduction of manufacturing costs. Further steps of improvement of the existing prototype concerning task specific enhanced use of aerostatic guiding and use of mechanical decoupling have been outlined. Due to the comparatively small mass of the stage very high dynamics are reached. Therefore the main application field can be deduced to be the highly efficient complete production of highly accurate parts with medium dimensions.

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ELASTIC ELEMENT SHOWING LOW STIFFNESS LOSS AT LARGE DEFLECTION

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ABSTRACT

Compliant mechanisms like a parallel guidance and a cross pivot flexure contain leaf-springs. The basic elastic elements like leaf-springs and wire flexures constrain Degrees-Of-Freedom (DOFs) when flat or straight respectively. However, when deflected these elastic elements lose stiffness rapidly. Therefore, mechanisms containing leaf-springs and wire flexures are suitable for small deflections only. This paper presents an elastic element, a so-called 'curved hinge flexure', which shows only a limited stiffness decrease at large rotational deflections of the initially 3 stiff directions. When comparing a curved hinge flexure with a single leaf-spring with comparable rotational compliance over a ± 15 degrees stroke, the stiffness in the stiff directions is 2.5 to 14 times higher.

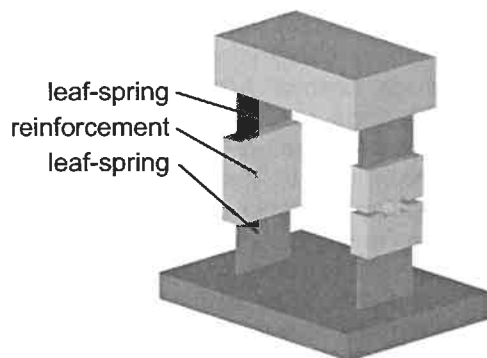


FIGURE 1. Exactly constrained parallel leaf-spring mechanism with reinforcements.

INTRODUCTION

In precision engineering, mechatronics and microelectromechanical systems, elastic elements are utilized for creating a deterministic behavior of the suspension of precision manipulators. With the principle of exact constraint design [1] systems often require a large suspension stiffness combined with a low

actuation stiffness. This ratio can be high if relatively long and thin wire-flexures or leaf-springs, the basic elastic elements, are used. A leaf-spring ideally constrains 3 DOFs and a wire flexure ideally constrains 1 DOF [2]. However, this is true for respectively a flat leaf-spring and a straight wire flexure. These types of flexures lose stiffness rapidly when deflected. Being built with the same basic elastic elements, mechanisms like a cross-pivot flexure [3], a parallel leaf-spring guidance [2] as shown in Figure 1, a folded flexure (often used in MEMS) and a folded leaf-spring [2] suffer a strong decrease in stiffness when deflected. Reinforcing the leaf-springs of the above-mentioned mechanisms, as shown in Figure 1 for a parallel leaf-spring mechanism, helps to increase the buckling load. A second option is to implement cross pivot flexures to increase the buckling load even further. However, neither option increases the stiffness of the shuttle significantly for large deflections.

STIFFNESS

To calculate the stiffness of a deflected leaf-spring, a bending moment is applied at the leaf-spring end in the rz -direction. The coordinate system is attached to the free end of the leaf-spring and rotates according to the possibly large rotation rz . At a certain rotation of interest $rz = Rz$, the coordinate system is fixed, and the stiffnesses are examined. The stiffnesses of the end of the leaf-spring can be defined as dividing an infinitesimal input force by the resulting displacement in a certain direction as shown in Figure 2. These 'stiffnesses' correspond to the reciprocal of the diagonal terms of the compliance matrix:

$$c_x = \left(\frac{\partial x}{\partial F_x} \right)^{-1}, c_y = \left(\frac{\partial y}{\partial F_y} \right)^{-1}, c_z = \left(\frac{\partial z}{\partial F_z} \right)^{-1}$$

$$k_x = \left(\frac{\partial r_x}{\partial M_x} \right)^{-1}, k_y = \left(\frac{\partial r_y}{\partial M_y} \right)^{-1}, k_z = \left(\frac{\partial r_z}{\partial M_z} \right)^{-1}$$

A rotating reference frame is chosen because we are interested in the compliance in the radial and tangential directions with respect to the center of rotation.

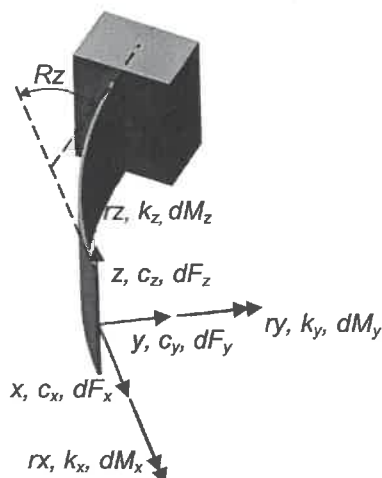


FIGURE 2. Deflected leaf-spring. The coordinate system is fixed at a certain angle Rz , after first traveling with the free end of the leaf-spring.

In order to keep the models simple with a limited number of degrees of freedom, the leaf springs are modeled as beams here using the multibody program SPACAR [4]. CosmosWorks finite element calculations have been made for verification. In the model shear is taken into account, but constrained warping and anti-elastic curvature are not taken into account.

LEAF-SPRING

A leaf-spring ideally constrains three degrees of freedom (DOFs) if it is flat. The three relatively stiff DOFs are the longitudinal elongation (x) and in-plane bending (z and ry) shown in Figure 2. The stiffness in twist (rx) and out-of-plane bending (y and rz) is relatively low.

The stiffness ratio k_y/k_z for example of a leaf-spring with dimensions $50 \times 50 \times 0.6$ mm at $Rz = 0$ is 6.9×10^3 , which shows the potential of the leaf-spring element to obtain a large suspension stiffness combined with a low actuation stiffness.

However, when deflecting the leaf-spring in one of the three compliant directions the stiffness of the three stiff DOFs decreases. For example when deflecting the 0.6 mm thick leaf-spring by

an angle of 15° in the Rz -direction, the stiffness ratio k_y/k_z has decreased by a factor 100 to 6.9×10^1 , due to the decrease in k_y . The stiffness as a function of the Rz -deflection of two leaf-springs in 6 DOFs is given in Figures 5 and 6. The stiffness ratios c_x/c_y and c_z/c_y also show strong decreases, 2.9×10^2 and 2.8×10^1 respectively, when deflected in Rz -direction.

CURVED HINGE FLEXURE

We present a so-called 'curved hinge flexure', shown in Figure 3, which shows only a limited stiffness decrease at large rotational deflections in Rz -direction. In terms of degrees of freedom the curved hinge flexure constrains three DOFs, x , z and ry and has one compliant DOF in the rz -direction. The DOFs y and rx are in between compliant and stiff. Therefore the curved hinge flexure is best used as a hinge flexure leaving compliant 1 DOF. An example application is the reinforced parallel leaf-spring mechanism shown in Figure 1, where the four leaf-springs can be replaced by four curved hinge flexures to obtain a higher stiffness at deflection.



FIGURE 3. The curved hinge flexure consists of two pre-curved leaf-springs. In this case the dimensions of the leaf-springs are $50 \times 50 \times 0.35$ mm.

In its basic form the curved hinge flexure consists of two curved leaf-springs in its undeflected, low stress state. In this undeflected state the leaf-springs are pre-curved and contribute both to, for example, the longitudinal stiffness. The longitudinal stiffness however is not at its maximum due to the curved shapes of the leaf-springs. When bending the curved hinge flexure in Rz -direction, one leaf-spring will straighten while the other becomes more curved as shown in Figure 4. The longitudinal stiffness of the more straightened leaf-spring will increase

faster than longitudinal stiffness of the more bended leaf-spring will decrease. Therefore the overall longitudinal stiffness will increase. The overall longitudinal stiffness increases until one of the leaf-springs is exactly straight, after which the overall longitudinal stiffness will drop. The stiffness change in the longitudinal direction applies in more or less the same way to the stiffness change of c_z and k_y .

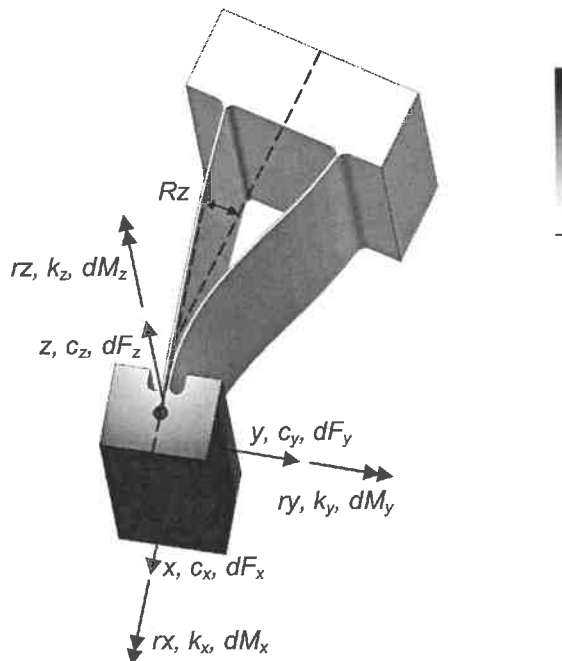


FIGURE 4. The pre-curvedness of the leaf-springs is optimized in such a way that if the suspended body is loaded by a bending moment around the z-axis, one of the leaf-springs becomes exactly straight, in this case at 10° Rz-rotation.

The basic idea behind the curved hinge flexure is to decrease stiffness loss at deflection, due to curving of initially flat or straight flexure elements, by incorporating curved flexure elements in the undeflected state which become straight when deflected. Parallel chains of elastic elements are required for maximizing the stiffness at different amounts of deflection.

For example, a curved hinge flexure has been optimized in such a way that during an increased external bending moment at a certain instant one of the leaf-springs becomes exactly straight at 10 degrees (Figure 5 and 6). The gap between the leaf-springs at the rotating side is 0.25 mm. The Young modulus is taken 200×10^9 N/m² and the Poisson ratio is taken 0.29. A curved hinge flexure, having leaf-spring

dimensions 50 x 50 x 0.35 mm, is compared with a single leaf-spring with dimensions 50 x 50 x 0.6 mm over a $\pm 15^\circ$ Rz-stroke because they show almost equal rotational stiffness k_z . The undeflected stiffness c_x , c_z and k_y of the curved hinge flexure is respectively 12, 6.7 and 40 times higher. However, the lowest stiffness c_x , c_z and k_y of the leaf-spring over a range from 0 to 15° is respectively 14, 3.9 and 2.5 times less than the curved hinge flexure. The stiffness loss at deflection of a leaf-spring can be minimized by taking a thicker leaf-spring, 1.25 mm for example. The worst case c_x , c_z and k_y -stiffness is 1.6, 0.49 and 0.29 times lower than that of the curved hinge flexure, however the k_z -stiffness is also 7.7 times higher.

The stress and strain of the curved hinge flexure needs to be taken into account. The strain of the curved hinge flexure in the example rises up to 0.30%, which can be a point of concern.

FUTURE OPTIMIZATION

The stresses and rz-stiffness of the presented curved hinge flexure can be minimized by optimizing the dimensions of the curved leaf-springs. An other idea is to use more than two parallel kinematic leaf-springs.

CONCLUSION

The curved hinge flexure shows less stiffness decrease at large deflections compared with a leaf-spring. When comparing a curved hinge flexure with a single leaf-spring with comparable rotational compliance over a $\pm 15^\circ$ stroke, the worst case stiffness in the stiff directions are 2.5 to 14 times higher.

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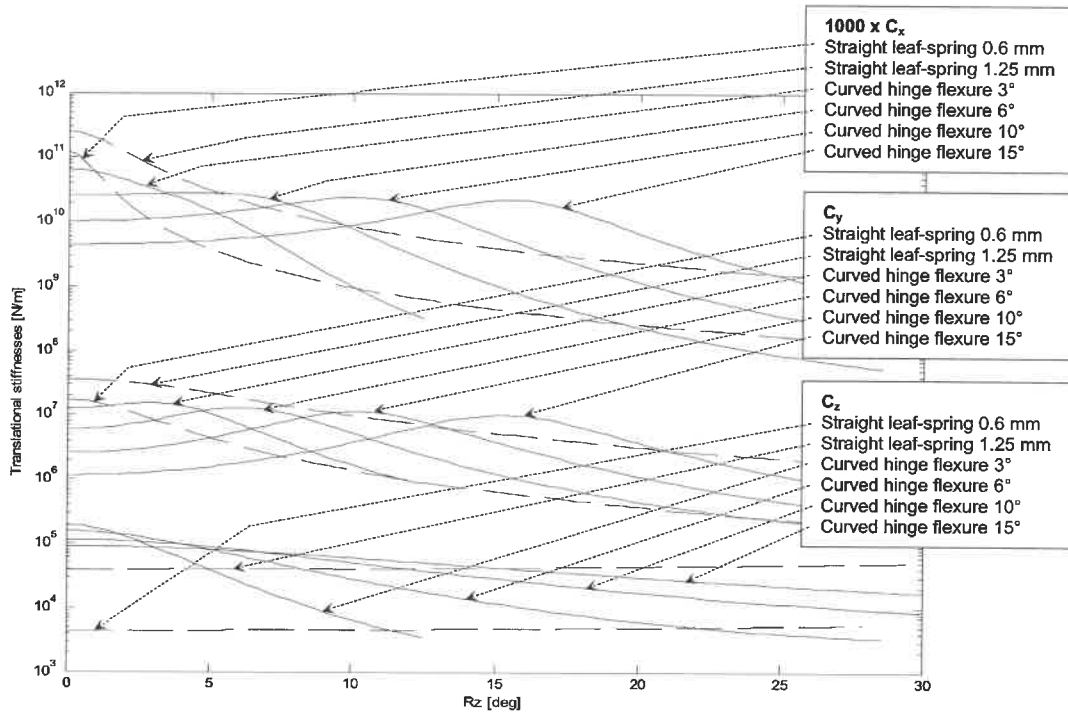


FIGURE 5. Graph showing the translational stiffnesses $1000 \times c_x$, c_y and c_z of two single initially straight leaf-springs with dimensions $50 \times 50 \times 0.6$ mm and $50 \times 50 \times 1.25$ mm, and four curved hinge flexures with each of the two leaf-springs having the dimensions $50 \times 50 \times 0.35$ mm. The four curved hinge flexures are optimized for maximum stiffness at 3°, 6°, 10° and 15° deflection in R_z .

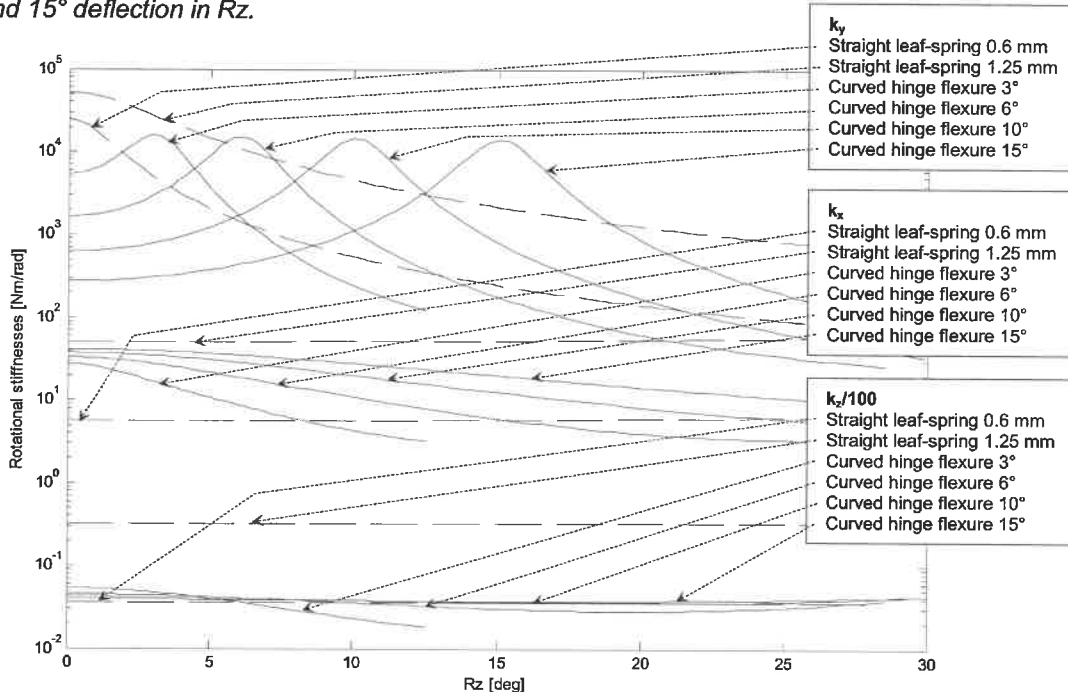


FIGURE 6. Graph showing the rotational stiffnesses k_x , k_y and $k_z/100$ of two single initially straight leaf-springs with dimensions $50 \times 50 \times 0.6$ mm³ and $50 \times 50 \times 1.25$ mm, and four curved hinge flexures with each of the two leaf-spring having the dimensions $50 \times 50 \times 0.35$ mm. The four curved hinge flexures are optimized for maximum stiffness at 3°, 6°, 10° and 15° deflection in R_z .

VOLUMETRIC ERROR MAPPING TECHNIQUES FOR THE NIST M48 COORDINATE MEASURING MACHINE

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The NIST Moore M48 coordinate measuring machine has provided NIST with unique and very flexible measurement capabilities for many years. Following the move of the machine into the new Advanced Measurement Laboratory (AML) on the Gaithersburg, MD site, the machine's state of the art performance was further improved through rigorous volumetric error mapping that takes advantage of the improved control and monitoring of environmental influences. The volumetric error mapping of the NIST M48 CMM follows mostly

traditional and historical guidelines however the accuracy requirements of the machine limit the usefulness of commonly use components such as angular and straightness optics. Attention is also given to the effects of the machine warm up behavior and the impact on the mapping and positioning stability. This paper will detail the equipment, optical systems, and procedures used to develop the full 21 parameter error compensation map used on each of NIST M48 CMM's.

REALIZATION OF ISARA 400: A LARGE MEASUREMENT VOLUME ULTRA-PRECISION CMM

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INTRODUCTION

In various fields of manufacturing and research, a growing demand for ultra-precise 3D measurement of large products exists. To fulfill this demand, IBS Precision Engineering has developed a new ultra-precision CMM with an unprecedented ratio of measurement volume vs. measurement accuracy. The design concept of this machine, the Isara 400, was presented at the ASPE spring topical meeting 2009 [1]. The current work presents various design details and the realization of the Isara 400.

ISARA HISTORY

In collaboration with Philips Applied Technologies, IBS PE previously developed the Isara CMM, an ultra-precision 3D Coordinate Measuring Machine, based on the design by Ruijl [2], with a measuring volume of 100 x 100 x 40 mm and a 1D measuring uncertainty of 15 nm, see figure 1.

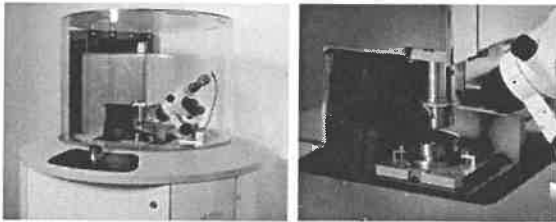


FIGURE 1. Photographs of the first Isara CMM.

In response to the demand for 3D measurement of larger products with features that require nanometer measuring uncertainty, the new Isara 400 ultra-precision CMM is developed.

ISARA 400 CONCEPT

The new Isara 400 CMM has a measurement volume of 400 x 400 x 100 mm. Three plane-mirror laser interferometers are applied as measuring systems for the machine axes. The interferometers each measure against the sides of a mirror table, on which the work piece is mounted. These interferometers are mounted in a single body metrology frame, which also holds the probe system. The laser beams are aligned to the probe tip and their mutual alignment does

not change during movement of the axes, thus fulfilling the Abbe principle [3] in 3D within the complete measuring volume.

In the configuration of the translation axes, a machine concept was chosen in which the variable product mass does not need to be moved vertically (figure 2). The product is mounted on the mirror table, which moves only in X- and Y-direction over a granite plate, guided by air bearings. The complete metrology frame moves in Z-direction, with guiding provided by air bearings against a vertical granite surface. This metrology frame contains the three laser interferometers and the probe, thus maintaining the Abbe alignment. The design of several key components is described in the next sections.

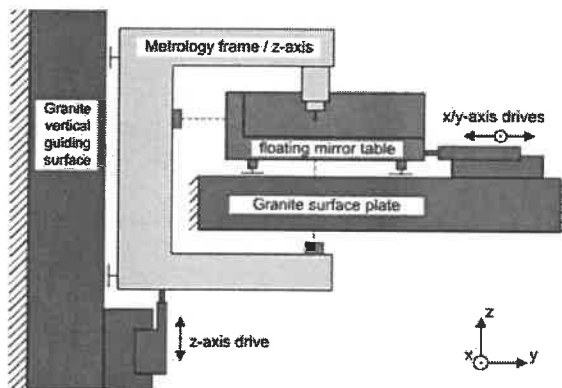


FIGURE 2. Concept of the Isara 400 CMM.

MIRROR TABLE

The mirror table of the Isara 400 is a monolithic Zerodur part with three reflective sides (figure 3). An additional product table is used as an interface between product and mirror table. This Silicon Carbide (SiC) product table was designed to be both light and stiff. The three supports of the product table are placed directly above the three supports of the mirror table, so that the weight of the product and product table does not cause additional deformation of the mirrors. The mirror table is supported by three flat air bearings, whose preload is provided by the weight of the mirror table assembly with product.

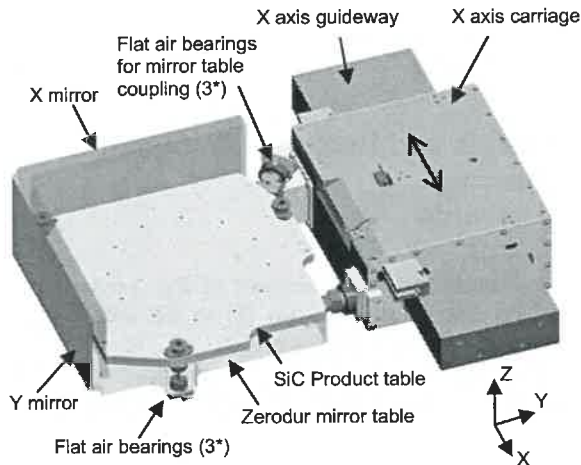


FIGURE 3. Floating mirror table and X-axis drive

The coupling between the floating mirror table and the X-carriage must define three degrees of freedom (X, Y and Rz) and allow for differences in thermal expansion. As the coupling is directly connected to the mirror table, it must be very compliant for parasitic motions between the mirror table and the X-carriage (Z, Rx and Ry): such parasitic motions should not cause deformations of the mirrors. Especially the flatness of the Z-mirror is sensitive to forces exerted by the coupling due to such motions. Parasitic motions may result from non-flatness and non-parallelism of the X-Y guide ways and sagging of the guideways because of the moving parts. Furthermore, the coupling must deal with different fly heights of the floating table resulting from different product loads. FEM calculations on the original coupling with three struts [1] show large deformations of the mirrors.

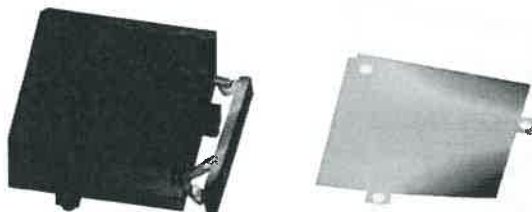


FIGURE 4. Effect of Ry rotation on coupling with three struts and resulting Z-mirror deformation.

As can be seen in figure 4, an applied Ry rotation on the X-carriage slide results in deformation of the Z-mirror. For example, the sagging of the X-axis guideway will cause a rotation Ry of 25 μ rad, which results in a mirror deformation of 200 nm.

Most of the parasitic motions will reproduce during the X and Y movements, so their

influence will be measured and corrected during calibration of the mirrors, which will be performed on the machine. Nonetheless, variable mirror deformations should be avoided. In the new design, shown in figure 3, the three struts are replaced by three air bearings. This configuration eliminates influences from vertical motions and also thermal effects will not lead to forces acting on the mirrors, because the frictionless coupling of the bearings cannot transmit any forces in Z direction. The parasitic Ry motions will only act on the two bearings at the back side, Rx motions will act on all three bearings. An error budget was made of all possible parasitic rotations and the worst case situation was used as input for a FEM analysis. With this new design, the effect on the flatness of the three mirrors was found to be < 1 nm.

As the three bearings do not move, the preloading can be achieved by adding extension springs directly to the bearings. These compliant springs will cause a force-closed preload for the bearings. The influence of those springs on the mirrors was also calculated and found negligible.

METROLOGY FRAME AND Z-DRIVE

The metrology frame, shown in figure 5, must maintain the mutual position and alignment of the probe and the three laser interferometers with high stability.

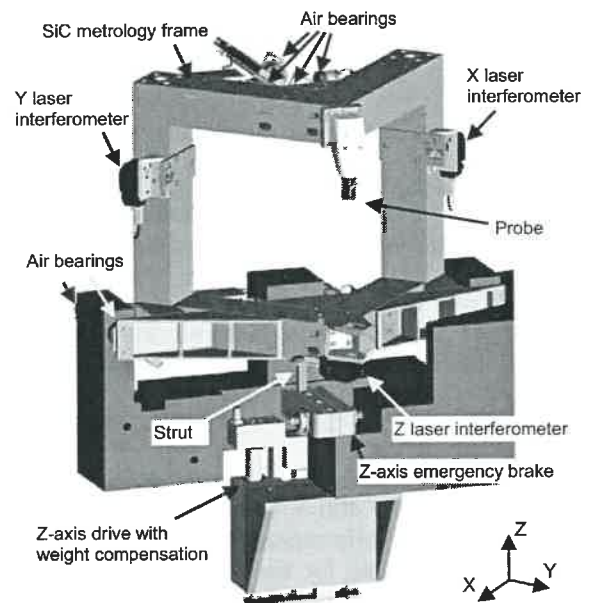


FIGURE 5. Metrology frame and Z-axis drive. Granite parts are only partially shown.

The metrology frame is statically determined in six degrees of freedom by means of five flat air

bearings and a strut joint to the Z-drive. All bearings are preloaded with flat air bearings on the opposite side. As the machine is a multi-probe machine, it contains a kinematic probe mount for a quick and reproducible exchange of probes. The metrology frame is an assembly of SiC beams; figure 6 shows several components.

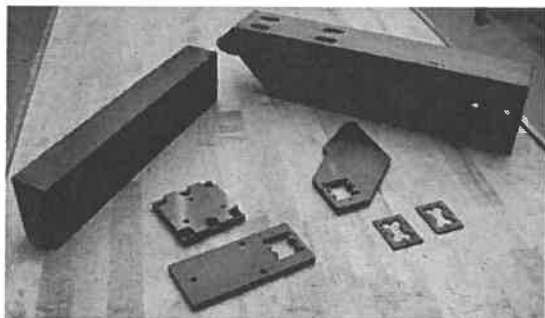


FIGURE 6: Photo of SiC metrology frame components (courtesy of Coorstek).

The Z-drive moves the metrology frame in vertical direction. It is placed below the metrology frame and connects with a strut directly through the centre of mass. Around the coupling strut an emergency brake is placed which can hold the metrology frame in place by clamping the strut, for example when no electric or pneumatic supply is present.

The Z-drive contains an air bearing guide for the linear motor, with an integrated weight compensation system, see figure 7.

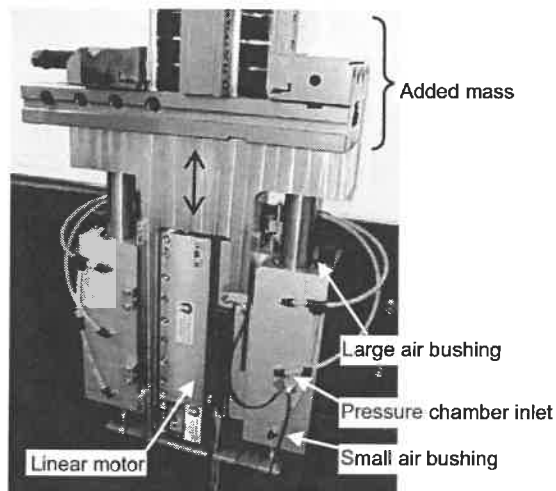


FIGURE 7. Photograph of realized Z-axis drive with weight compensation system.

This bearing system consists of two shafts; each shaft is guided in two cylindrical air bearings (bushings) with different diameters. The space

between the two bushings is sealed by the two bushings, thus serving as a pressure chamber. By supplying exactly the right air pressure, the weight of the metrology frame can be compensated and equilibrium is achieved. As a result, the required force from the linear motor to hold the metrology frame in place is minimized. The heat generation of this drive is therefore very low. The realized Z-drive shown in figure 7 is currently being tested.

TRISKELION ULTRA-PRECISION PROBE

Conventional touch probes are generally not suitable for ultra-precision coordinate metrology, due to their relatively high measuring uncertainty and large probing forces and the ability to measure small features is limited, due to the relatively large probe tips which are applied.

Several different ultra-precision touch probes developed over the past years [4,5,6,7] share a common design feature: the stylus is elastically suspended, allowing deflection of the probe tip and thus reducing probing forces. This deflection is measured by means of an integrated measurement system within the probe. Building on these previous efforts in the development of ultra-precision touch probes, the new 'Triskelion' probe system was developed (figure 8).

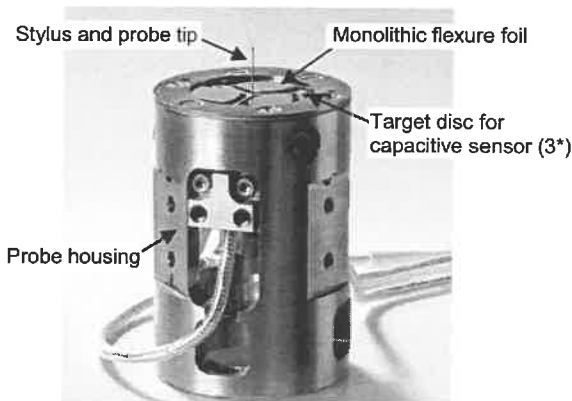


FIGURE 8. Triskelion ultra-precision probe.

The probe calibration was described in [1]. Absolute measurement errors are < 10 nm per axis of the coordinate system and < 15 nm in 3D.

Several product measurement have been performed, using the new probe system integrated into an ultra-precision CMM. Figure 9 shows a photograph of the measurement of an aspherical mould insert and the corresponding form error of the profile measurement across the center, which is obtained after subtraction of the theoretical asphere.

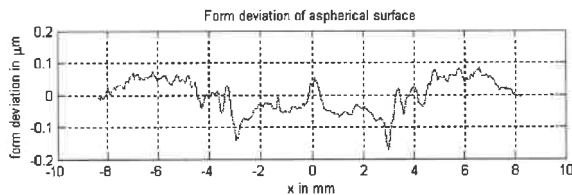


FIGURE 9. Aspheric mould measurement using the Triskelion probe.

MEASURING UNCERTAINTY OF ISARA 400

An extensive uncertainty analysis was performed to make an accurate estimation of the expected measuring uncertainty of the Isara 400 CMM. Some of the most important contributing factors are the uncertainty of the laser interferometers, the calibration uncertainty of the mirror table (flatness and perpendicularity), the stability of the metrology frame and the measurement uncertainty of the probe. For all axes, the expected 1D measuring uncertainty is 45 nm ($k=2$), whereas the full-stroke 3D measuring uncertainty totals 100 nm ($k=2$). Both values include contributions from the described probe system and are valid within the complete measuring volume of the machine.

CONCLUSION

The new Isara 400 CMM is the latest development of IBS Precision Engineering for coordinate metrology of large, complex parts with nanometer level measuring uncertainty. The expected 3D measuring uncertainty is 100 nm within the complete measuring volume of 400 x 400 x 100 mm. Tactile probes, such as the presented Triskelion ultra precision touch probe, as well as other possible (optical) probe systems can be used to perform scanning or point measurements. The presented machine, see figure 10, provides a technology basis which can be adapted and optimized for specific user requirements. The machine will be operational by the end of 2009.

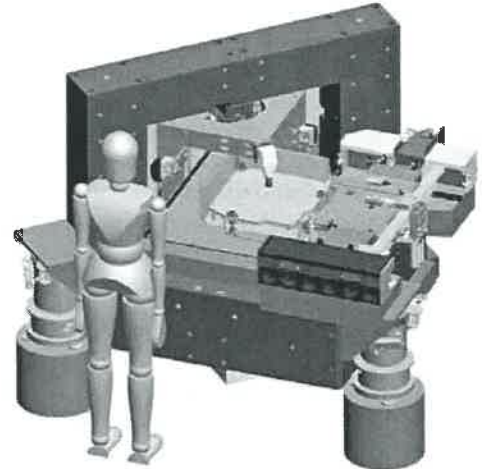


FIGURE 10. Overview of the Isara 400 CMM.

The work presented in this document is part of the European sixth framework programme projects *NanoCMM*, FP6-026717-2 and *Production4µ*, FP6-2004-NI-4.

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THE MEASUREMENT OF DIMENSIONAL METROLOGY

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INTRODUCTION

Dimensional metrology drives precision engineering. Since precision engineering and metrology are inexorably linked; insight into the commoditization of precision engineering can be gained through a study of the dimensional metrology market. Tracking Average Sell Price (ASP) over time indicates commoditization occurs at a standard rate.

METHODOLOGY

Dimensional metrology is the measurement of length. Metrology in general spans seven fundamental units^[1]: Meter – unit of length, kilogram – unit of mass, second – unit of time, ampere – unit of electrical charge, Kelvin – unit of thermodynamic temperature, Mole – unit of amount of substance, and candela – unit of luminous intensity. It is well known that to accurately measure length other metrology parameters must be known or controlled, for instance temperature. These other metrology parameters are auxiliary; therefore all the other measurement types will be ignored in this study.

Economic Data Sources

The core economic data was gathered through surveying the commercial producers of dimensional metrology systems. A seed group of companies was identified through interviews with experts in the field. This seed group was followed into competitive companies and complementary technologies, leading to a comprehensive survey.

Data sources included annual reports, 10K documents, product brochures and announcements, internet sites, Google, and personal interviews with companies, and users. Data timing is fixed at 2007 values due to the timeliness of reports.

Over 190 companies and major subsidiaries were catalogued along with over 500 products. It is estimated that this represents over 80% of the market in identified companies and over 90% of the total revenue.

Database

A database was developed on sales revenue and average sell price for each identified product. The data was structured on technology, measurand, and sales per market.

Measurand

The dimensional metrology market was separated into seven measurand as shown in Table 1.

Measurand	Technologies
Distance/ Angle	Glass scales, lasers, capacitive and inductive gages
3D Cloud	Coordinate Measuring Machines
AOI	Automated Optical Inspection: vision systems and microscopes
Film Thickness	Ellipsometer, Interferometer, Spectrophotometers,
2D Profile	Stylus, AFM, single point probes
3D Profile	confocal, interferometer, chromatic
3D Form	Interferometers

TABLE 1. Measurand and surveyed technologies

Markets

Markets reported include only industrial markets. The markets selected were: Semiconductor Wafer, Semi Packaging, Flat Panel Displays, MEMS, Precision Machining, Photovoltaics, Optics/Photonics, Magnetic Data Storage, and Optical Data Storage. Each of these markets contains the practice of precision engineering. Medical metrology such as retina thickness metrology was not included.

MARKET DATA

The dimensional metrology market in 2007 was \$8,700M USD with an uncertainty of ~10%. As seen in Figure 1. 65% of the market is found in Precision Machining (automotive, aerospace, and medical implants for instance) with ~20% in Semiconductor Wafer metrology. This is surprising since common wisdom has

Semiconductor is a much bigger metrology market at over \$5,000M in annual revenues in 2007. The confusion arises between wafer INSPECTION, identifying the presence of defects or contamination, and wafer metrology, such as measuring the flatness of a wafer or transistor gate critical dimension (CD). This study only reports wafer metrology. Interestingly, even including inspection the Precision Machining metrology market is larger.

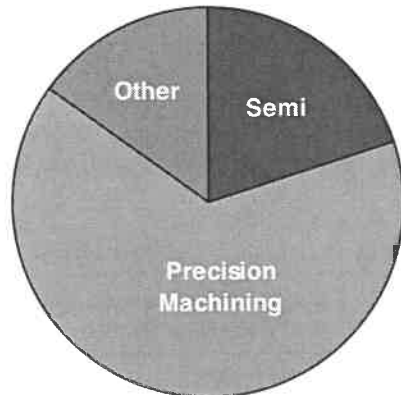


FIGURE 1. Precision Machining dominates the dimensional metrology market, followed by Semiconductor Wafer metrology

Market Segment Size by Measurand

Distance/Angle and 3D Cloud dominate the dimensional metrology market. This is not surprising as distance is the fundamental unit and therefore the primary single measurement desired.

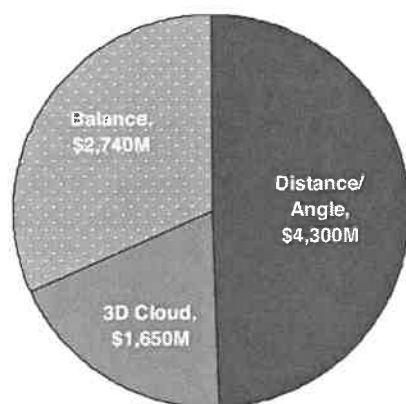


FIGURE 2. Distance/Angle and 3D Cloud dominate the dimensional market. Balance is comprised of the remaining five measurand

The "Balance" of the market is nearly equally shared by the remaining five measurand,

Automated Optical Inspection (AOI), Film Thickness, 2D Profile, 3D Profile, and 3D Form.

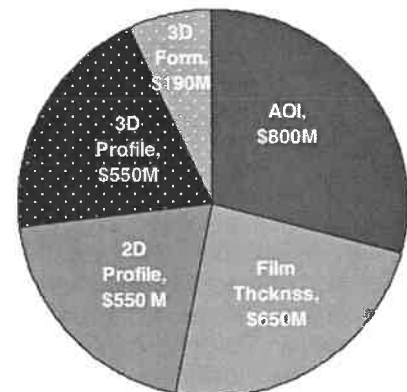


FIGURE 3. The "balance" of the market was \$2,740M USD in 2007

Average Sell Price

A key parameter that differentiates these data is average sell price (ASP). ASP for each measurand was estimated by averaging the list price of all products in each measurand category. As seen in Figure 3, the ASP has large variation across the measurand surveyed.

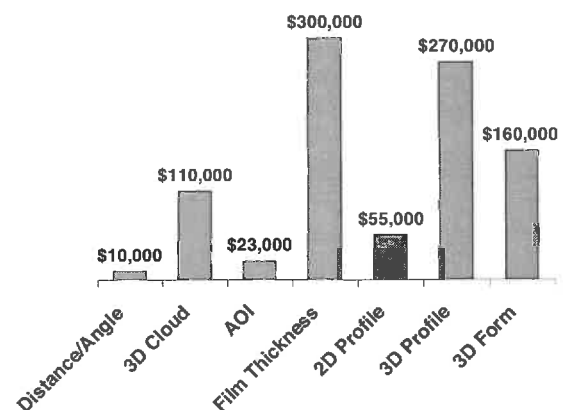


FIGURE 4. Average Sell Price (ASP) for surveyed measurand. Film Thickness and 3D Profile clearly dominate

Does Application Drive ASP?

Semiconductor applications dominate Film thickness and 3D Form and both have high ASP's. This is expected as wafer handling and clean room enclosures are very expensive and drive up ASP. Precision Machining applications dominate Distance and Angle and 3D Cloud which both have low ASP. Precision Machining applications tend to be much simpler systems with lower ASP. Not following this trend is 3D profile which has a high ASP but has no

dominant market or application and serves both semiconductor as well as precision machining markets. Therefore it is difficult to conclude that the main driver to ASP is application.

COMMODITIZATION OF METROLOGY

Commoditization can be defined as “a competitive environment where:

- Product differentiation is difficult
- Customer loyalty and brand values are low
- Competition is based primarily on price
- Sustainable advantage come from cost (and sometimes quality) leadership”^[2]

Price, as seen above, is a critical factor when considering commoditization. Therefore another approach is to investigate if there are any other drivers on price besides application.

One possibility is a relationship between a technology’s time on the market and ASP. Table 2 lists the measurand and estimated time-since-introduction. These are rough estimates with the uncertainty growing worse with the earlier technologies. Modern Distance/Angle tools are assumed to have been introduced at the beginning of mass production in the 1920’s. Stylus profilers are assumed to have been introduced in the early 1940’s. Recent developments are easier. 3D Profile was introduced in the early 1980’s and 3D Form in the early 1970’s.

Measurand	Time-Since-Introduction
Distance/ Angle	~90 years
3D Cloud	50 years
AOI	50 years
Film Thickness	20 years
2D Profile	~70 years
3D Profile	25 years
3D Form	40 years

TABLE 2. Measurand and estimated-time-since introduction

Commoditization Curve

Plotting the ASP data found in Figure 4 against the Time-Since-Introduction in Table 2 produces the scatter plot in Figure 5. When plotted on a log linear plot a linear relationship is found.

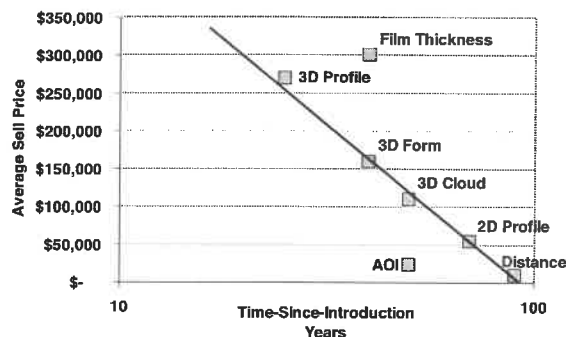


FIGURE 5. Time since introduction compared to ASP, has a linear relationship on a log-linear scale

Caution must be exercised regarding the quality of fit due the uncertainty in both ASP values and Time-Since-Introduction, but it is a compelling trend. Note Film Thickness sales are primarily in the Semiconductor market which drives ASP very high.

Commoditization Rate

The fit indicates that the ASP is halved every 20 years. This translates into a 3.3% price decrease per year in stable, non-inflated currency.

A test example of this idea is the 3D profiler. In 1984 the Wyko TOPO 3D, a Phase Measuring Microscope Profiler sold for ~\$60,000. Today an equivalent system, the Zygo NewView 600p, sells for \$50,000. With an average inflation rate from 1984 till 2009 of 3.1%^[3] the TOPO 3D would sell today for \$130,000. Applying the 3.3% commoditization rate yields an estimated sell of \$57,000. Definitely within the uncertainty of the data and suggestive the concept is sound.

FUTURE POTENTIAL RESEARCH

ASP research of historic metrology equipment like stylus profilers and CMM would prove interesting. By tracking their ASP change over time the notion of a commoditization rule for precision systems could be tested and if shown to exist, refined.

CONCLUSION

The metrology market offers insight into the commoditization of precision engineering and suggests precision engineering systems commoditize at a rate of 3.3% per year. Adjusted for inflation

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THE SPHERICAL PROFILOMETER POLARIS 3D

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INTRODUCTION

An instrument for the metrology of high-aspect surfaces has been constructed. This new profilometer has a spherical measurement volume 100 mm in diameter inside of which highly non-rotationally symmetric optical and mechanical surfaces with large surface sags can be measured with an accuracy of 100 nm. Aspheres with large departures, torics, biconics and freeform surfaces can be measured with deviations from a sphere of several millimeters.

By mounting a linear axis and an optical probe on top of a rotary table and rotating a surface on an orthogonal spindle a hemispheric measuring volume is obtained. An additional linear axis allows a variety of convex and concave surfaces to be measured while keeping the probe nearly normal to the surface as it sweeps out a spiral path from the equator to the pole.

Challenges in the construction of this instrument include the alignment of the axes, measurement of their error motions, development of an appropriate compensation strategy for those errors and building a control system that is both user-friendly and fast enough to measure a part in less than two minutes.

MACHINE DESIGN AND CONTROL

A polar geometry measuring machine developed in by the Precision Engineering Center in 2000 [1, 2] has been upgraded with an additional rotary axis and linear axis to create a 3D spherical coordinate measuring instrument. The 2D Polaris machine used an air bearing LVDT as a probe. The new instrument uses an optical probe to rapidly measure a surface without damage. Shown in Figure 1 are two opposing sets of stacked axes; linear on top of rotary with a vertical axis of rotation (R and θ) and rotary on top of linear with a horizontal axis of rotation (Z and ϕ). The measuring probe is mounted on the linear R axis. As a part is rotated in a vertical plane by the second set of axes, arcuate motion of the probe in an orthogonal horizontal plane

allows a spiral set of data to be collected and assembled into a 3D surface profile.

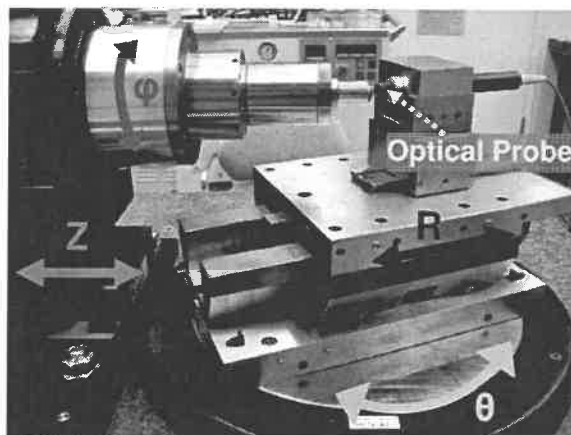


FIGURE 1. Polaris 3D.

All four of the Polaris 3D axes are air bearing with brushless motors and high resolution encoder feedback. Parts are held on the spindle with a collet chuck. The machine is controller with a Delta Tau UMAC which also collects measurement data. Subsequent analysis is performed on a separate computer using Matlab.

Optical Probe

Polaris 3D uses a STIL CHR 150 optical probe to avoid damage to finished optical surfaces. In this probe an axially chromatic lens and double spatial filter is used to produce a confocal imaging system for distance measurement. For any probe lens to object distance, light of a specific wavelength (i.e., color) will be in focus. The CHR 150 has a spot size of 8 μm and a maximum measurement range of 300 μm at a working distance of 5 mm. Its axial accuracy is 90 nm and lateral resolution is 4 μm . The steepest slope that can be measured from a specular surface is 25°, although this value is highly dependent on material and surface characteristics. The output of the probe controller is an analog signal linearly scaled into the adjustable measurement range. A second analog output is proportional to the intensity of

the measurement signal. The probe bandwidth is 1 kHz. The probe pen can be seen in Figure 1 mounted on the R axis.

The measurement signal voltage increases as the distance from the probe to the surface increases. The intensity signal represents the "quality" of the reflected light and decreases as the probe is moved closer to the sample or as the slope is increased. A highly specular surface may saturate the detector or there may not be enough returned light from a diffuse or rough surface. Both the intensity and displacement output voltages are zero when the probe is too close or too far from a reflective surface or if there is too much reflected light or not enough reflected light. Changes in specularity, scratches and sharp slope changes may all result in the loss of a signal from the probe.

Surface Tracking with Probe Feedback

It is desirable to use the probe not only as a measurement device, but for the machine controller to use the probe as a secondary feedback source such that the machine axes "follow" the shape of the part. This enables the measurement of complex surfaces without programming the machine axes to move in a trajectory specific to each part.

The R axis has two operating modes. It can be commanded to move independently of the other machine axes either from the user interface or with manual controls. Alternatively the servo control loop for R can consider the optical probe measurement signal as an error that should be driven to zero since the probe moves with the R axis. This type of feedback loop is easily implemented in the machine controller. The control action for this follower servo loop is intentionally weak so that the R axis is stable. This algorithm is depicted in Figure 2. A rate limiter prevents axis oscillation in response to analog noise and the control filter ceases to update the input to the integrator if the probe signal is not "good". Thus the R axis will complete the accumulated command and simply stop moving if the probe signal is lost. In the worst case of a rapidly changing probe signal that moves through its full range in one servo cycle the R axis will move no more than 150 μm trying to respond to the probe. In practice, the R axis position lags a rapidly changing probe signal by about 20 μm and never moves more than twice that amount when the probe signal is

lost. This isn't a problem for a measuring instrument since the surface displacement is a combination of the Z, R and probe displacement signals as a function of the θ and ϕ positions. Less than 20% of the probe range is used during a measurement with tracking engaged.

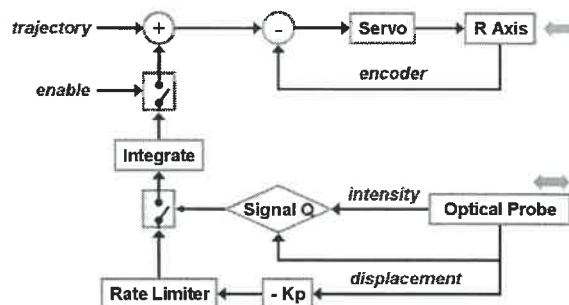


FIGURE 2. Probe Tracking Algorithm.

GEOMETRIC ERRORS

Construction of Polaris 3D required multiple alignment procedures and error compensation schemes. Geometric errors such as spindle errors of rotary axes and straightness, roll, pitch and yaw of the linear axes were all considered. In addition to these axis errors, the relative alignment of the axes was critical to establishing the spherical coordinate system on which all measurements would be based. Certain error motions were found not to contribute, others could be made insignificant through careful adjustment of the axis alignments and still others are compensated for based on independent measurements of the axis errors.

Alignment Errors

One example of an error that could not be adjusted to a satisfactory level was the alignment of the rotational axes θ and ϕ of the machine. Ideally, the θ and ϕ axes would intersect at the origin of the spherical coordinate system. The screw adjusters for making the adjustment only allow reliable adjustment to within 2 μm . The residual error is shown schematically in Figure 3. Since this error contributes directly to measurement error and at 90° contributes 100% to measurement error, it has subsequently been measured independently and a compensation scheme has been devised. Figure 4 shows the residual error when measuring a calibration sphere from +90° to -90° in θ . The slope of the cones is equivalent to the difference $\Delta R(x)$ between the two cones as a function of radial distance from the intersection point of the two cones. If m is the slope of the

residual cone, the alignment error can be calculated as,

$$e_{x\phi} = \frac{\Delta R(x)}{2m}$$

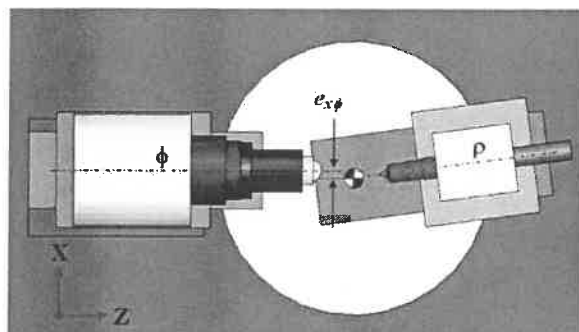


FIGURE 3. The centerlines of the θ and ϕ - axes do not intersect and are separated by a distance of $e_{x\phi}$ in the x-direction.

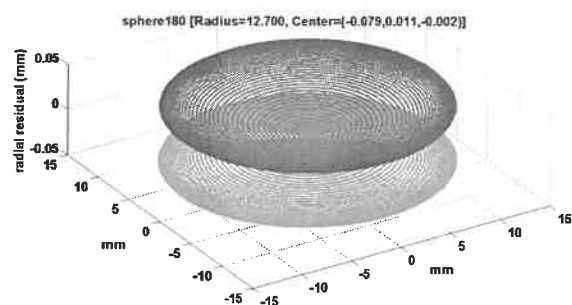


FIGURE 4. Residual of best-fit sphere to measurement from $+90^\circ$ to -90° in θ .

Given the measured alignment error, new coordinates for the R and θ axis positions can now be found:

$$R' = \sqrt{R^2 + e_{x\phi}^2 + R e_{x\phi} \sin \theta}$$

$$\theta' = \cos^{-1} \left(\frac{R'^2 - R^2 + e_{x\phi}^2}{R' e_{x\phi}} \right)$$

Probe Alignment Errors

There are three probe alignment errors: ρ , τ and γ that adversely impact measurement. The first two have been described extensively for the 2D version of Polaris [3]. The γ -error is new to the 3D version and represents the misalignment of the probe relative to the ϕ -axis spindle centerline. Measurement of this error is achieved using a tilted flat mounted on the ϕ -axis and acquiring probe data at $\theta=0^\circ$ as the ϕ -axis rotates. Ideally, with the probe perfectly

centered, the output of the probe would not change. Any misalignment produces a sinusoidal output proportional to the radial probe displacement from the spindle axis. The amplitude of the sine wave divided by the tangent of the artifact's tilt angle gives the magnitude of the misalignment while the phase of the alignment gives the contribution in the x- and y-directions. Shown in Figure 5 is data over five revolutions with the y-axis aligned with $\phi=0^\circ$.

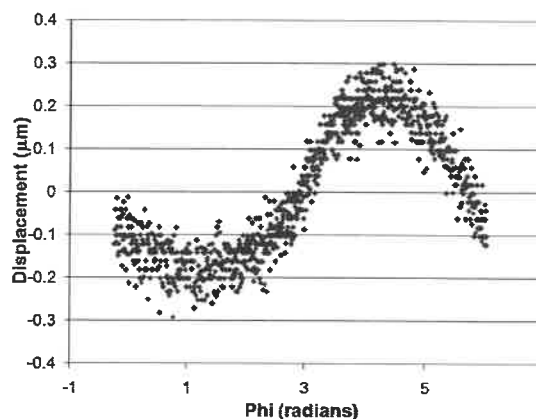


FIGURE 5. Tilted flat measurement for probe y-alignment. Note the mean value of the data at zero radians. This value is proportional to the probe alignment y-error.

Given a tilt angle for the flat of 7.4° , the y-error is thus $-0.77 \mu\text{m}$ and the x-error is $-1.5 \mu\text{m}$. The x-error in this measurement is incidentally equivalent to the τ -error and thus offers an independent means of verifying this alignment as well.

MEASUREMENT RESULTS

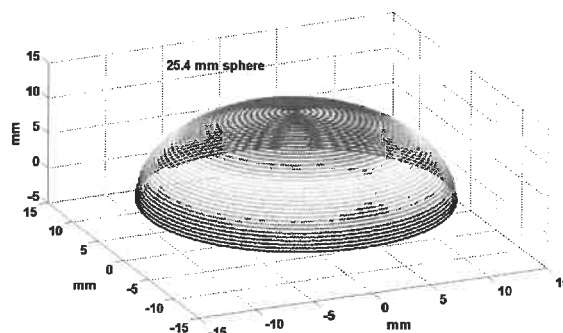


FIGURE 6. Calibration Sphere Measurement.

Calibration Sphere

A 25.4mm diameter Bal-Tec sphere was measured 15 times to assess repeatability and

accuracy. One of the measurements is shown in Figure 6. The mounting of the sphere on the spindle had significant runout: approximately 80 μm . For a sphere (or any rotationally symmetric surface) the runout can be easily removed from the resulting data.

The radius of the best fit sphere to this data is 12.7002 mm. The standard deviation in radius for the repeated measurements is about 100 nm.

Non-rotationally Symmetric Part

A non-rotationally symmetric convex surface measured at a coarse spacing is shown in Figure 7. Asymmetric features are revealed in Figure 8 after a best-fit sphere is removed from the data. The residuals in the radial direction from the center of this least-squares sphere are plotted.

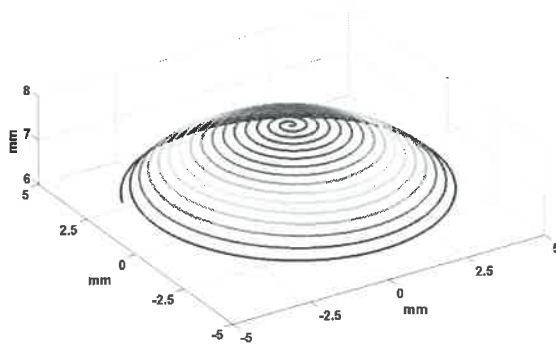


FIGURE 7. Non-rotationally Symmetric Part.

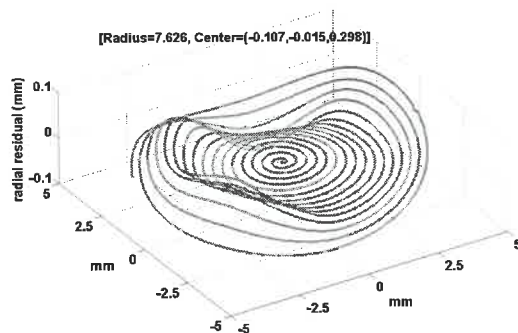


FIGURE 8. Radial Residuals.

CONCLUSIONS

The construction of Polaris3D has been completed, the axes aligned and the control and data acquisition system is operational. Matlab

code for assembling the raw axis encoder and probe signal data into spherical coordinates is complete. Codes for comparison of a measurement with a representation of the intended part shape are being developed. Polaris3D will ultimately represent a comprehensive system for sophisticated measurement and analysis of optical and mechanical parts with several millimeters of aspheric or non-rotationally symmetric departure from a spherical profile.

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FULL FIVE DEGREE-OF-FREEDOM ERROR CHARACTERIZATION OF ULTRA-PRECISION SPINDLES USING THE MULTIPROBE METHOD

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ABSTRACT

This paper demonstrates an implementation of five degree-of-freedom precision spindle metrology at high speed in a production environment using a multiprobe method. A single probe is used with a purpose-built fixture, enabling measurement of axial, radial and tilt spindle error motion components as defined by the *ASME B89.3.4 Standard: Axes of Rotation; Methods for Specifying and Testing* [1]. This multiprobe error separation method can be more convenient to perform while still providing results as good as or better than those obtained through reversal.

INTRODUCTION

The ultra-precision air bearing spindle shown in Figure 1 is produced for diamond-turning and grinding applications. The demanding accuracy requirements for this spindle could only be certified through design of a new generation of metrology tooling. The contribution of typical artifact form error to the total measurement exceeds the spindle error specification. Furthermore, if not properly separated, artifact errors can mask spindle errors—such as 2-lobed ball errors cancelling 2-lobed spindle errors. As a result, the nanometer-level artifact form error must be separated from spindle error motion using an error separation technique.



FIGURE 1. A Professional Instruments Company ISO 5.5 motorized air bearing spindle.

Although the theory and application of error separation is widely known, this work addresses the challenges of hardware design when testing at operating speed. The reversal techniques described by Donaldson and Estler theoretically provide the simplest method for separating spindle error motion from artifact form error [2,3]. However, in a production environment, measurement accuracy of these techniques suffers from the difficulty of exact artifact reversal. Particular care must be taken to accurately remount the artifact exactly 180° without influencing form error, which can be difficult when testing at high speed. Any change in the artifact between the tests, for example a fingerprint or dust, is split evenly between the spindle and artifact. In addition, the probe must be positioned along an identical artifact measurement track after indexing.

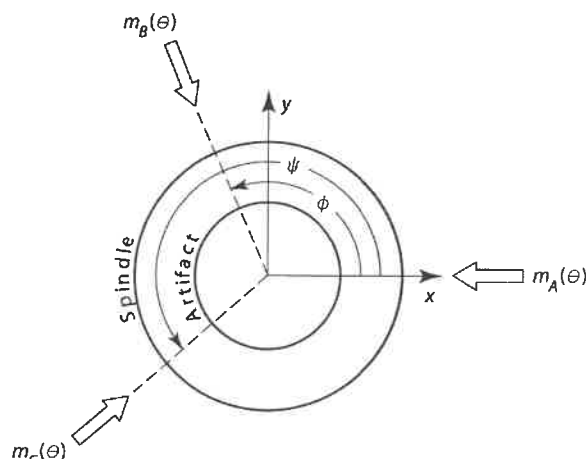


FIGURE 2. Multiprobe error separation method applied to radial and face measurements.

This implementation of the multiprobe method assesses a single component of synchronous error motion from three asymmetrically orientated measurements. The sensor is moved from 0° to 99.844° and 202.5° without indexing the artifact. While this method does not perfectly separate all of the components of spindle and

artifact error, the chosen orientation angles minimize harmonic distortion.

Spindle error motion is characterized by five components—one axial (Z), two radial (R_x and R_y) and two tilt (α_x and α_y). Three measurements are required to obtain each radial and tilt component. Tilt error motion components are computed from two radial measurements at different axial locations (150 mm separation). The axial component of spindle error does not require error separation.

APPROACH

The spindle under test in this work (Professional Instruments ISO 5.5) features a 1024-count rotary encoder (Renco R22i) and a brushless, frameless motor and amplifier (MCS custom-wound motor and MCS AX-2000 amplifier). The capacitive sensor (Lion Precision C23-C, 0.5 nm/mV) targets a $\varnothing 25$ mm lapped sphere as shown in Figures 3 and 4. The amplifier (Lion Precision CPL290) incorporates a 15 kHz first-order, low-pass analog filter with linear phase response. The data acquisition system (Lion Precision SEA V8.3) is triggered by the encoder index pulse, providing immunity to synchronization errors caused by speed variation. Further low-pass digital filtering is done in software.

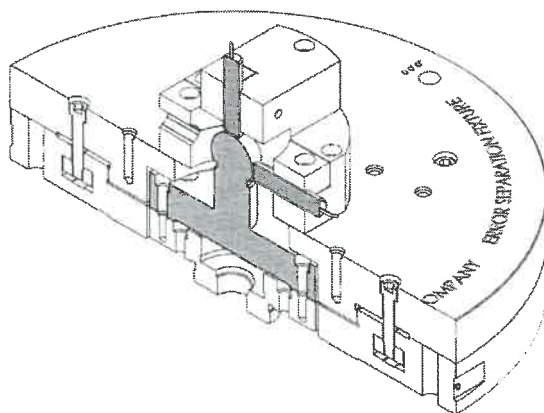


FIGURE 3. Radial multiprobe error separation tooling with $\varnothing 25$ mm lapped spherical artifact.

Several potential contributors to measurement uncertainty are eliminated by sequentially indexing a single sensor through the three orientation angles. These locations are provided by jig-ground holes in the stator-mounted index plate shown in Figure 5. Accurate repositioning of the sensor avoids the problems with mismatched sensors, but at the expense of simultaneous data collection.



FIGURE 4. Precision $\varnothing 25$ mm lapped spherical artifact for spindle error motion testing.

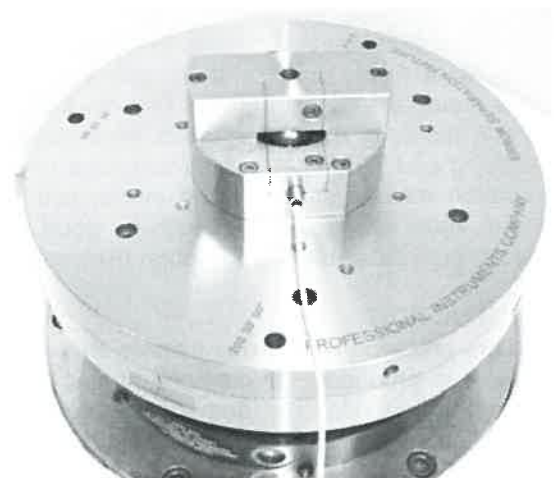


FIGURE 5. Tooling for radial multiprobe error separation method.

The design of the hardware is consistent with Moore's requirements of an inspection tool [4].

- It is accurate: The accuracy is based on sensor calibration and probe location.
- It requires a minimum of operator skill: This tooling significantly simplifies the procedure for spindle testing, resulting in improved efficiency and accuracy because it does not require repositioning of the artifact.
- It inspects a specific type of error: The tooling enables measurement of radial, axial and tilt synchronous error motions at known locations.
- It is fast to use: Elimination of the need to unbolt the artifact from the spindle and rotate it 180° significantly decreases the total time to complete a test.
- It is self checking: Each spindle measurement also re-checks the artifact. Furthermore, the index plate can be rotated to different angles to check repeatability. In addition, the tooling is designed to allow Donaldson reversal.

RESULTS

Previous work has described various iterations and improvements made to the radial error motion tooling shown in Figure 5 [5]. This tooling was designed to allow separation of artifact and spindle errors with multiprobe separation and Donaldson's reversal. Multiprobe separation is used with the sensors located at 0°, 99.844° and 202.5°. The sensor indexing plate is then rotated 90° to measure the orthogonal component of radial error and the multiprobe separation procedure is repeated.

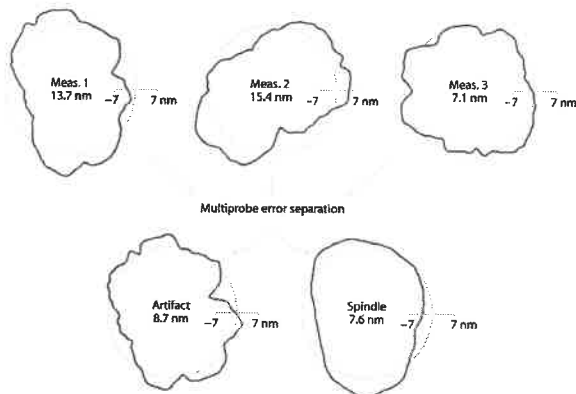


FIGURE 6. Radial multiprobe error separation result at 500 RPM.

Typical synchronous radial error separation results are shown in Figure 6. The measurements recorded at the three orientation angles are a combination of spindle error and artifact form error. A mathematical manipulation of the three measurements enables the separation into 7.6 nm spindle error and 8.7 nm artifact form error.

As mentioned previously, the axial component of spindle error does not require error separation. The artifact's imperfections do not influence the axial measurement except by second-order effects. Total synchronous axial component is

0.4 nm fundamental and 0.9 nm residual as shown in the measurement in Figure 7.

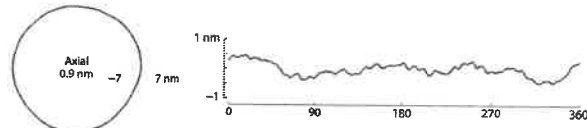


FIGURE 7. Residual axial spindle error motion at 500 RPM. The fundamental axial is 0.4 nm.

The tilt error motions shown in Figure 8 are often of greater concern than radial error motions when diamond-turning optics. The tilt error motion at a particular radius when combined with the axial error motion results in the face spindle error motion. We perform the tilt calculation in the X and Y directions using two radial measurements separated by 150 mm in the axial direction (low and high).

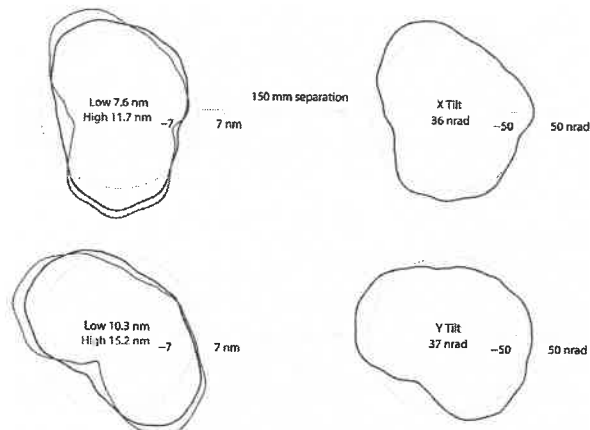


FIGURE 8. Tilt multiprobe error separation result for X tilt and Y tilt (α_x and α_y) at 500 RPM.

The five degree-of-freedom characterization is complete with the combination of Figures 7 (axial) and Figure 8 (two radial and two tilt).

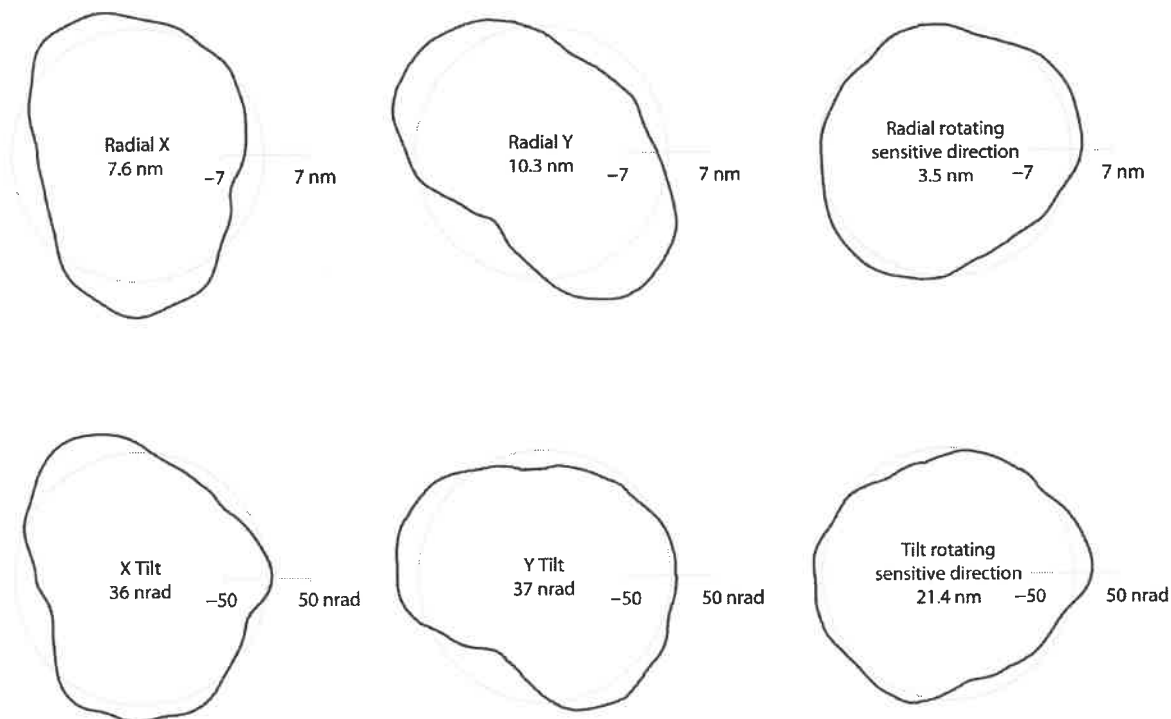


FIGURE 9. Results for two radial and two tilt degrees-of-freedom at 500 RPM.

CONCLUSION

Five degree-of-freedom spindle error motion measurement at the nanometer-level is demonstrated in a production environment. The multiprobe error separation method used here eliminates the need to index the artifact which simplifies the testing procedure.

Measurements are made sequentially with a single probe to reduce certain error sources. The tilt error motion is calculated from the radial multiprobe separation result at two axial distances.

These results confirm that multiprobe error separation can be a reasonable alternative to reversal methods in a production environment requiring testing at high speeds. Our next generation of spindle testing tooling will enable the measurement of rotating sensitive face and radial error motions in a single fixture.

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DESIGN OF A PRECISION TRANSFER LINE FOR DIP-PEN NANOLITHOGRAPHY

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SYNOPSIS

In this paper we describe a precision indexing system that was designed as a scalable, probe-based fabrication system for high-rate, mass nanomanufacturing. This system is roughly the size of a 'bread box', and contains a precision belt-drive that has been designed to enable continuous processing of substrates as dies or continuous ribbons/webs. This approach enables nanofabrication over a large area without the need for large, expensive, high-speed precision motion stages. Here we bring the substrate to/from the processing area via 'web transport.' We envision this technology will (a) provide a means to transform probe-based nanofabrication processes into practical high-volume nano-manufacturing processes and (b) enable parallel probe-based processing of biological specimens (e.g. stem cells) during factorial experiments.

INTRODUCTION

In probe-based nanomanufacturing (PBN) micro-scale probe tips are used to make, manipulate, or measure nm-scale features on a substrate. Two important issues for PBN are cost appropriate equipment that can (i) deliver the desired rate and (ii) process large (1000s cm^2 or more) substrate surface areas. The rate

issue is being addressed by the use of 2D probe arrays that consist of up to 55,000 tips [1]. This leads to the need for 6 axis positioning equipment that can position the probes and level the array with respect to the substrate surface. The 'processed area' issue has yet to be resolved due mainly to the perception that the solution is in old paradigms, e.g. equipment that travel over large surfaces with nm-level resolution. In this approach, achieving suitable cost with six-axis nanopositioning is difficult.

In this paper we describe the design and early experimental characterization of a prototype precision indexing system that combines a six-axis flexure system, an array of probe tips and precision belt-drive. The system is shown in Fig. 1 and its main components are shown in Fig. 2. This combination enables (i) continuous fabrication – moving long webs/ribbons of a substrate material within a nanofabrication processing center, or (ii) semi-continuous fabrication – moving arrays of dies/chips atop a belt "transfer line" that moves dies/chips within a nanofabrication processing center. The system is designed to meet the needs of general PBN processes that require high rate and large area processing. The utility of the system is best described in the context of a specific process, therefore we will use dip pen nanolithography (DPN) as the example.

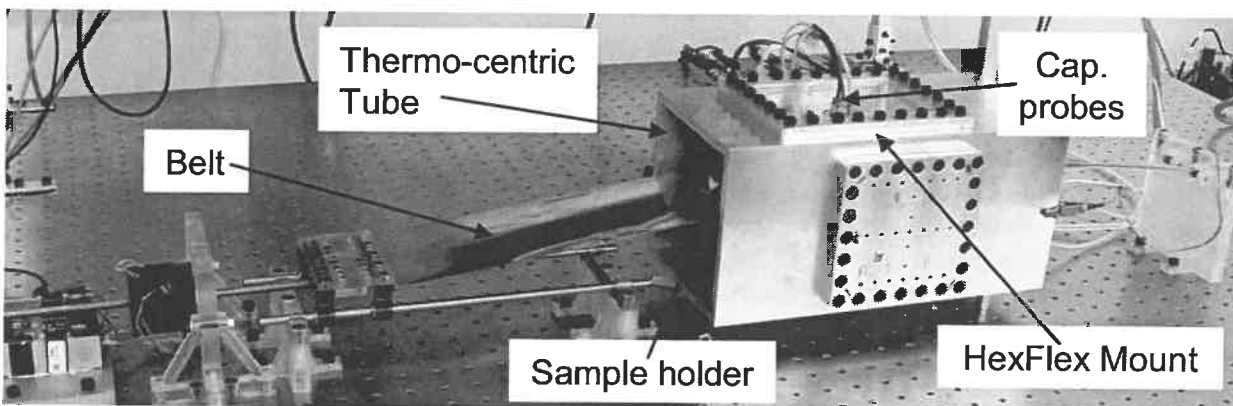


FIGURE 1: Prototype DPN transfer line and HexFlex-based processing center.

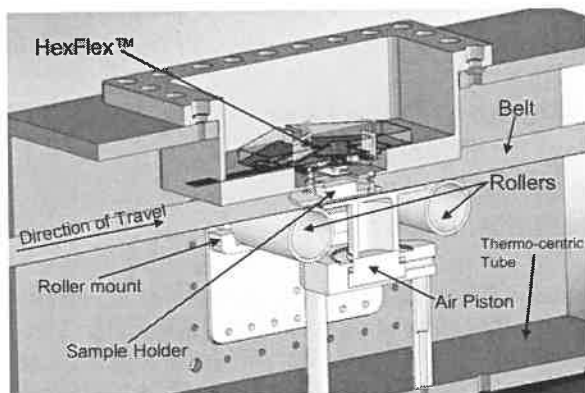


FIGURE 2. Main elements of prototype system and material flow within the system

Overview of DPN processing

DPN is a PBM process wherein a material, e.g. ink, is deposited onto substrates with feature sizes down to 15nm [2]. The "ink" is written upon a substrate via an AFM-like tip. Figure 3 shows how this process happens [3]. As the tip is dragged across a substrate, ink diffuses from the tip to the substrate via a nano-scale water meniscus. The meniscus forms spontaneously via capillary condensation [4].

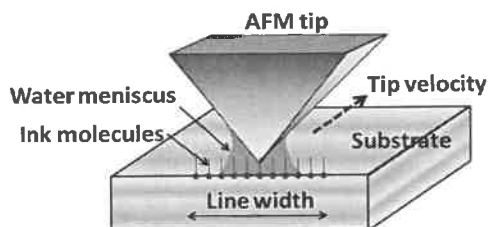


FIGURE 3. Detail of DPN process

DPN tip arrays, as large as 50,000 tips over 1cm², may be used to process many samples on individual die (dies normally consist of an array of 50,000 samples over 1 cm²). With existing equipment, it is not possible to process large surfaces (1000 cm² or more) at reasonable cost.

Case study

DPN, and most other PBN processes, are viable fabrication processes, but a means to process many dies and/or large area substrates is needed for industry application. Our case study is based upon a need expressed by the Ohio State's Center for Affordable Nanoengineering of Polymer Biomedical Devices. They create arrays of DNA bundles for subsequent processing by writing of chemical agents upon

the bundles. Each bundle of DNA contains approximately ten strands, and each strand of weighs approximately 5.23×10^{-20} kg. At best, meaning if processing yield is 100%, each time a 50,000 probe tip array writes upon a DNA array it is processing approximately 29 pg of DNA. If processing of arrays occurs at 0.1 Hz, then a single array has a 2.9 pg/sec processing rate. This logic is carried out to provide estimates for production rate in Table 1.

TABLE 1. DNA production rate

Required DNA	100	µg
Number of tips/array	55000	Tips
Number of arrays	1	Trays
Number of arrays	1	Samples
Cycling frequency	0.1	Hz
Best case yield	100	%
Processing time	402.4	Days
Production rate	0.25	µg/day

The purpose of processing the DNA is personal drug development, e.g. custom drugs for genetic diseases, 402 days is clearly too long. The processing center has been designed for high throughput and reduced processing time.

EQUIPMENT FUNCTIONS AND PROCESSING

The transfer line provides long-range indexing of substrate material into, and out of, the system. Material flow and processing steps are best understood via Fig. 4. The substrates are indexed beneath a 1cm² array of probe tips that are attached to a meso-scale HexFlex [5]. A flexible linear scale is affixed to the edge of the belt and follows the belt through the system. A read head records the motion of the belt, thereby providing the means for fine resolution position control of belt/web indexing.

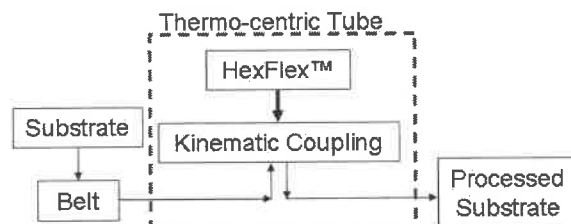


FIGURE 4. Prototype system material flow.

An air piston, positioned below HexFlex station, presses the belt and substrate up to the HexFlex. The substrate material (die or web) contains kinematic alignment features (grooves)

that engage with mating features (balls) which are attached to the HexFlex mount. The kinematic features align the substrate to the HexFlex within ± 300 nm ($0.3\mu\text{m}$). The HexFlex is a six-axis flexure that performs fine alignment (nm-level position and μradian parallelism beyond the KC's performance) between the probe tip array and substrate. The HexFlex has a range of motion of $50\mu\text{m} \times 50\mu\text{m} \times 50\mu\text{m}$. It then puts the array in contact with the substrate and moves the probe tips across the substrate's surface during processing. When processing of the local area is complete, the transport system indexes a new substrate below the HexFlex, the position/orientation of the new substrate is registered, and the cycle repeats.

DESIGN

The housing and structure of the prototype was constructed through the use of a square aluminum extrusion. This design is used to reduce complexity/part count, and isolate the processing zone from the environment. A large hole in the top of the tube receives and houses the HexFlex nanopositioner sub-system. The two holes in the side receive inserts that contain the rollers' alignment flexures.

Quasi-kinematic couplings [6, 7] are used to ensure precision alignment (e.g. during tool changes or maintenance) and provide sealing contact between the inserts and the tube. The rollers, which support and drive the web/belt, are assembled between the side inserts. Sets of kinematic couplings are used to align the substrate to the HexFlex, the HexFlex to its insert, and this insert to the structural tube. The HexFlex-substrate KC has a capture distance of ~ 250 microns. Technically, the indexing system need only position the substrate to within this distance to permit engagement of the coupling, but coupling repeatability improves when the balls-grooves are better aligned prior to mating.

This tubular architecture was selected as this design may be scaled to include other parts of a nanomanufacturing system. We envision that several segments of tubes (the prototype segment is approximately 30 cm long) may be coupled together by kinematic couplings. Each segment would be specifically designed for a different fabrication process. The connected segments would form a low-cost, modular, flexible 'manufacturing line', wherein the environment in each segment may be controlled.

COUPLING PERFORMANCE

We have already demonstrated processing capability with the HexFlex sub-system [5]; therefore we focus here on the performance of the new elements: the indexing system and substrate-HexFlex kinematic mounting system. For a typical probe-based manufacturing system, the plane of probes and substrate plane must be aligned within a few μrads to set the spacing of the substrate and probes to avoid crashing or too large a gap between probe and substrate. In DPN, the probe tips rest on compliant cantilevers and the process is meant to be conducted with tips in contact. The leveling constraints here, $800\mu\text{rad}$, are less stringent.

Three capacitance probes were used to measure the leveling repeatability of the substrate-HexFlex coupling. Figure 5 shows the probes within the device and Fig. 6 shows the positions of the probes relative to the substrate-HexFlex kinematic coupling.

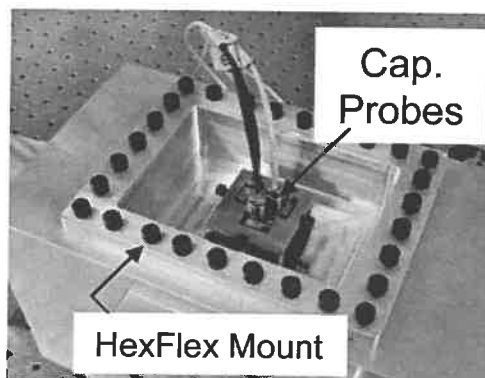


FIGURE 5: Measurement setup

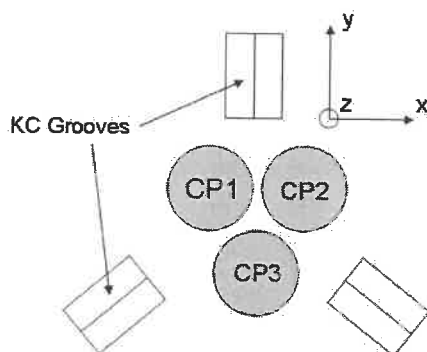


FIGURE 6: Capacitance probes position relative to kinematic coupling grooves.

The substrate was brought to the processing zone via the indexing system (in the $+\hat{y}$

direction). The substrate was then preloaded onto its kinematic mounts via the air piston, and then the system was allowed to settle.

The repeatability data are plotted in Figures 7 and 8. The repeatability in z was ± 300 nm. This is within the HexFlex's vertical range of motion ($12\text{ }\mu\text{m}$) and better than the z alignment of $6\text{ }\mu\text{m}$ required for the probe tips. The initial transient position in Fig. 7 is indicative of a coupling wear-in period and was expected. The repeatability was therefore calculated using the stabilized readings, i.e. cycles 11–40.

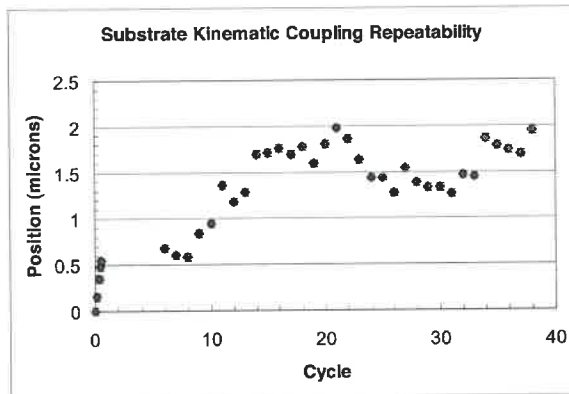


FIGURE 7. Z axis repeatability

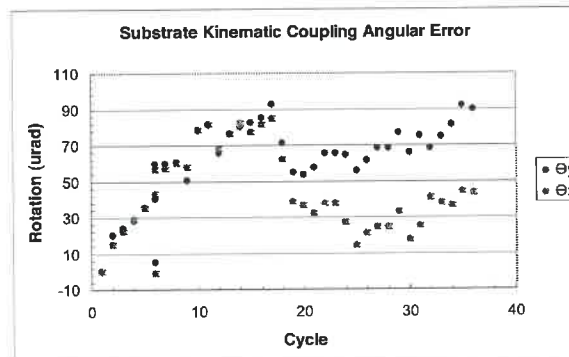


FIGURE 8. Tip and tilt repeatability (error bars are smaller than data points)

The repeatability in tip-tilt, shown in Fig. 7, was $\pm 20\text{ }\mu\text{rads}$. This is 40 times better than the requirements of DPN but on the order of performance that most probe-based processes will require. The contrast between required and achieved performance is listed in Table 2.

TABLE 2. Required/measured performance

Parameter	Required	Measured
Vertical offset (z)	$\pm 6\text{ }\mu\text{m}$	$\pm 300\text{ nm}$
Angular error (θ_x, θ_y)	$\pm 800\text{ }\mu\text{rad}$	$\pm 20\text{ }\mu\text{rad}$

CONCLUSION

These experiments are the first conducted on the prototype. We envision further optimization of the system (e.g. improved friction management via flexures [8, 9]) will improve repeatability by a factor of two or more. Our next steps encompass the integration of many 55,000 tip arrays to increase throughput.

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MANUFACTURING OF FREEFORMS WITH WELL DEFINED REFERENCE STRUCTURES

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INTRODUCTION

Modern freeform designs are pushing the performance of optical systems used for beam shaping, beam homogenization as well as for imaging optics [1, 2]. However, without a tight relation between the optical and the mechanical coordinate system, even the most sophisticated freeform can not tap its full potential within the optical system. Therefore the manufacturing of references in close tolerances to the optical freeform surface is a big challenge. Diamond machining is an excellent technology to meet this requirement.

The present paper summarizes realized machine setups, freeform manufacturing processes as well as the cutting results for diamond machining of optical surfaces and reference structures. The results and investigations are based on practical examples.

WHY DO FREEFORMS NEED REFERENCE STRUCTURES

Classical rotationally symmetrical optics need tolerances for figure deviation, surface roughness, cleanliness ect. and for the position of the optical surface in respect to their mechanical boundaries. Well developed deterministic manufacturing techniques (SPTD, grinding, polishing) provide excellent optical surfaces. Nevertheless, without well defined mechanical references, high performance optics can not provide their full potential. In fact rotational symmetrical optics needs references for centering, tilt and vertex height. Over the years specific technologies were developed to push the accuracy for centering and vertex height in the micron and sub-micron range [3]. If the optical element is made of a diamond turnable material, references and the optical surface can be directly turned.

This issue is much more complicated in case of freeform optics because the references must represent six degrees of freedom. The challenges of freeform manufacturing like:

- Freeform measurement
- Refeeding the measurement data into correction loops
- Controlled assembly and adjustment strategies
- Repeatable positioning / clamping during manufacturing and assembly

are feasible only with well defined fiducials.

MANUFACTURING OF FREEFORMS AND REFERENCES BY DIAMOND MACHINING

Diamond machinable materials have the advantage, that all precise geometries, including the optical surface, references and respectively the mechanical boundary can be manufactured on one machine. However, in most cases several manufacturing techniques like servo turning, milling or shaping are necessary to produce the references and the freeform surfaces. One promising approach is to realize multiple cutting techniques in the same machine setup.

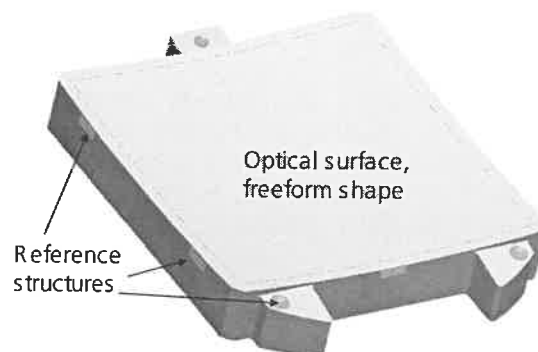


FIGURE 1. Design example of an optical freeform element with reference structures.

Combining Shaping and Turning

Shaping – known as a typical ultraprecise process for micro structuring – offers an excellent alternative for freeform manufacturing. Especially steep surfaces or freeforms without rotational symmetry are preferred shapes for 3D shaping. Using the machine setup, illustrated in figure 2, both shaping and diamond turning are possible in one setup and with one and the same tool. The machining technique is based on a four axis ultraprecision machine. During the shaping process the Y and Z axis are moving synchronously along the calculated tool path. The X - axis works in raster respectively positioning mode. The Y axis is equipped with a diamond tool. Measurement probes can also be mounted on to the Y – axis to reference the workpiece according to the machine coordinate system. The rotational C-axis allows the positioning of the workpiece with high precision in Rz or can act as a turning spindle.

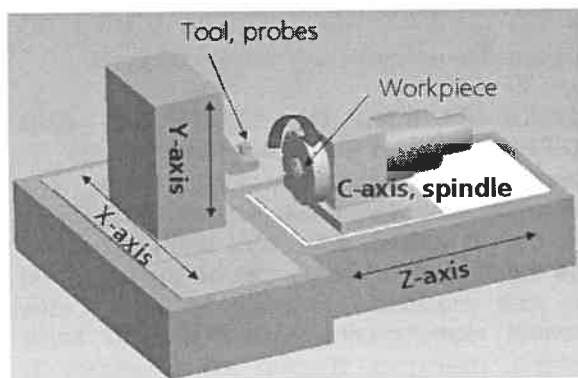


FIGURE 2. Machine setup for shaping and turning.

The setup was used for optical elements with one freeform side and one rotationally symmetrical side (figure 3). Due to the use of one tool, both optical surfaces have a tight positioning tolerance. A rotational symmetrical reference and mounting structure was also turned. Beside this application it is also possible to manufacture a freeform by servo turning and adding references by shaping.

This approach is typical for polymer optic prototypes. In this case the prototype consists of one rotationally symmetrical element and one freeform element. The optical axes of both surfaces are located in the spindle centre. Therefore the alignment and the referencing of the part are easy, because the tool can be aligned in a similar way to SPDT. The Tool

position (0, 0, 0) represents the origin of the optical coordinate system. The optical surface description can be used directly for creating the machining data.

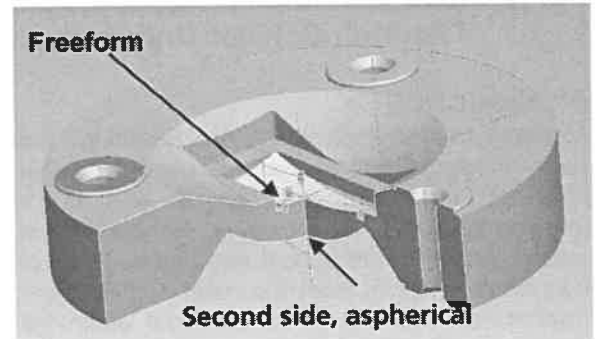


FIGURE 3. Freeform element, inserted into a rotationally symmetrical fixture which is used for: positioning, referencing and for a stress free mounting. The freeform element will be separated after diamond machining.

Combining C-Axis Milling and Turning

Slow-Tool machining offers a good way to manufacture freeforms that can be separated into a rotationally symmetrical and a freeform component with moderate slope angles. An integrated C-axis milling process is a potential solution for the manufacturing of reference structures in a close relation to the optical surface.

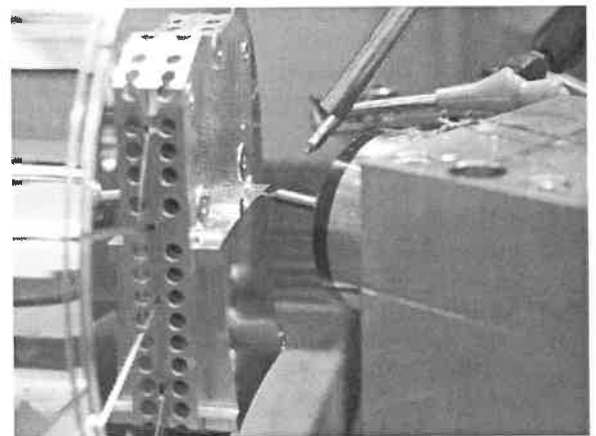


FIGURE 4. Milling additional reference marks next to a diamond turned off-axis mirror on an ultra precision lathe.

The arrangement of references must be capable of determining the position of the optical surface in all degrees of freedom. Therefore three

references on a common pitch circle or more elements on different diameters must be machined in a fixed position on the substrate to represent the axis of rotation. These alignment marks embody the coordinate system of the lathe and the manufacturing process of the part. It can be used to transfer the information about the position of an optical surface into the metrology. This allows avoiding the usual necessary alignment fit of the measurement data respectively the target surface in 6 degrees of freedom to calculate the form error of the surface.

The reference marks must be capable of being detected within sub- μm range during the measurement. Spherical references with steep slopes are suitable for the position detection with tactile measurement devices like 2½ D profilometers. But the slopes are limited using the Slow-tool machining. Therefore an additional milling spindle can be mounted next to the turning tool in the UP-lathe.

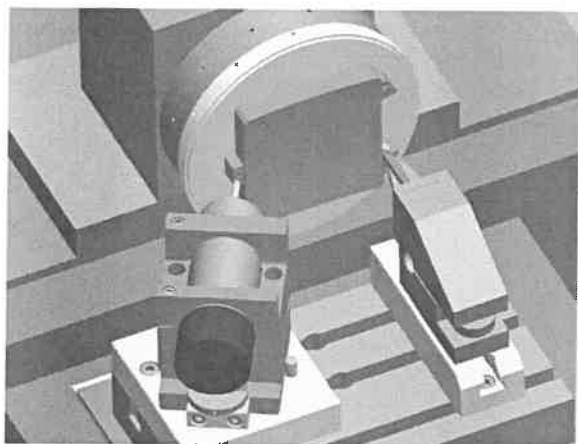


Figure 5. Basic Setup for combined C-Axis-Milling and Turning: Spindle/C-Axis with freeform work piece, Turning tool, additional milling spindle.

Essential for such a Slow-Tool servo setup is the proper alignment of the milling tool to the C-axis and the turning diamond. The tool must be aligned within sub- μm range to the axis of the turning spindle. A careful tool setting is achieved by milling specialized test geometries to analyze the tool decentration in each direction independently. After a precise tool setting, the tool can be aligned within a micron to the axis of rotation of the C-axis.

But also the programming of the tool path is challenging. Due to the off-axis position of the reference elements on the substrate, its

geometry has to be treaded as a freeform [4]. The machine kinematics of the UP-lathe consists of two linear axes (X, Z) and one rotational C-axis.

A proprietary software solution had to be programmed for the generation of the tool path.

The trajectory of the tool center point follows an Archimedean spiral in an off-axis position to cut the reference spheres. The path is generated by adding the normal vector of the surface with the length of the tool radius to the position vector of the node from the origin of the coordinate system.

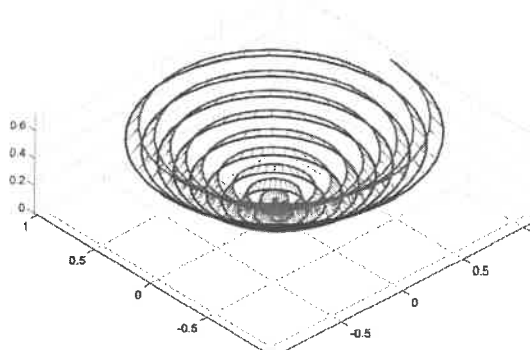


FIGURE 6. Toolpath trajectory for the c-axis milling derived from the contact point of the tool with the surface.

A following shift to the desired position on the substrate and a coordinate transformation leads to the point cloud that can be interpreted by the machine using a smoothing spline interpolation. The resulting machine motion is a synchronous oscillation of the C-axis and the X-axis, while the material is removed by the mono-crystalline diamond tool.

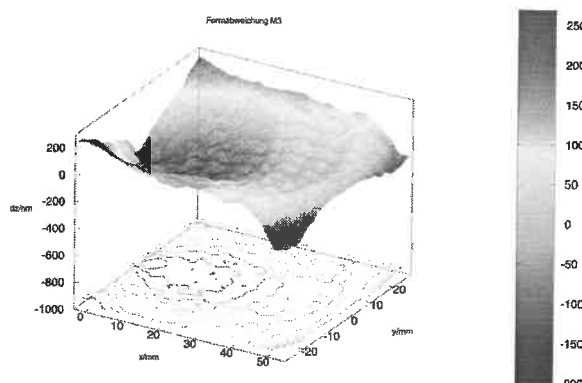


FIGURE 6. Form error of an off-axis mirror measured with a profilometer and aligned according to milled reference marks. Shape irregularities 380 nm p.-v. inside the quality area.

The 2½D profilometer Panasonic-UA3P offers the ability to measure the reference marks and the optical surface in a high accuracy regarding form and position.

The reference marks, which embody the coordinate system of the UP-lathe, are used to find the quality area of the optical surface and the tilt about the X, Y, and Z axes. A subsequent alignment step according to the measured angles results in the true form error of the surface.

This approach is especially useful for geometries which do not contain a vertex, like off axis mirrors and freeform mirrors.

CONCLUSION

The combination of different diamond machining techniques in one machine setup opens up the flexible manufacturing of freeform optics with well defined reference structures. One key issue of freeform manufacturing – getting confident measurement data and refeeding it in the correction loop – can be solved. Therefore mechanical references or optical fiducials are machined in sub-micron relation to the optical surface. Especially in the field of high performance mirrors for telescopes the excellent positioning of the optical surfaces are of particular importance. The proposed techniques enable for a deterministic and precise mirror mounting and system assembly process.

ACKNOWLEDGEMENT

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MEMS KINEMATIC CLAMP FOR REPEATABLE PRECISION ALIGNMENT

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INTRODUCTION

This work presents the design and fabrication of micro-scale kinematic clamps for Microelectromechanical System (MEMS) applications. Kinematic clamps have been used for myriad macro-scale engineering applications where precise and repeatable positioning is required; however, micro-scale kinematic clamps and passive precision alignment are relatively new and underutilized in MEMS. Conventional methods of wafer alignment involve optical inspection, which is not highly precise and is often time consuming. In contrast, with MEMS-scale Kinematic Clamps sub-micron alignment repeatability and reduced setup times for alignment are possible.

Previous research has investigated the precision performance of using a MEMS-scale elastic averaging technique [1]; however, the use of a purely kinematic clamp is under explored. Figure 1 demonstrates the implementation of the MEMS-scale kinematic clamp in aligning two silicon wafers. A MEMS kinematic clamp will enable technologies where precise alignment is required between dies or wafers such as optical MEMS, wafer and die-level bonding, and microfluidic lab-on-a-chip applications [2]. This paper discusses the design and fabrication of a "three-vee" MEMS-scale kinematic clamp for standard (100) silicon wafers.

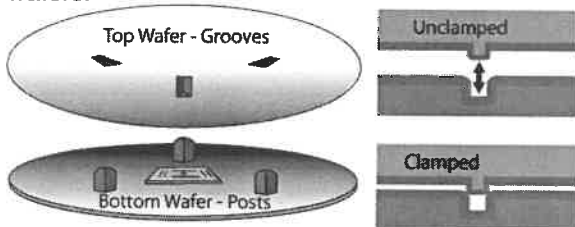


FIGURE 1. One wafer consists of grooves while the other consists of rounded posts. This technology enables fast and precise alignment for multi-wafer MEMS.

Properly designed kinematic clamps are deterministic; they only make contact at a number of points equal to the number of degrees of freedom (DOF) that are to be constrained [3]. To fully constrain a fixture, three vee-groove and post pairs can be utilized (constraining 2DOF per pair). This paper begins with a review of potential microfabrication methods for various vee-grooves and post designs. The most promising methods based on fabrication feasibility were chosen for further analysis to determine their theoretical precision capabilities, and specific fabrication challenges.

FABRICATION METHODS

Traditional kinematic clamp consist of balls and grooves, arranged in triangular configuration, which mate to form three sets of kinematic couples. The six points of contact fully constrain the two halves of the clamp. Several bulk and surface micromachining methods for fabricating the clamp's components were investigated. The major concerns for MEMS Kinematic Clamps are the surface roughness of the clamp's contacting surfaces and the tolerances of the tools which are used in the fabrication of the clamp. These factors are responsible for introducing positional errors in the clamp and reducing the clamp's overall precision, and they will be discussed along with the fabrication processes.

Vee-Groove Fabrication

One possible method for creating vee-grooves in silicon is through anisotropic wet etching. Potassium hydroxide (KOH) exhibits a high etching selectivity between the $\langle 100 \rangle$ and $\langle 111 \rangle$ crystal planes of silicon. The $\langle 100 \rangle$ plane etches at a rate 400 times greater than that of the $\langle 111 \rangle$ plane [4]. This preferential etching allows for the creation of vee-grooves with sidewall angles of 54.7° , as shown in Figure 2. The vee-groove depth can be set by calculating the dimensions of the rectangular mask openings.

For this process, a silicon nitride or silicon oxide masking layer is deposited across the silicon

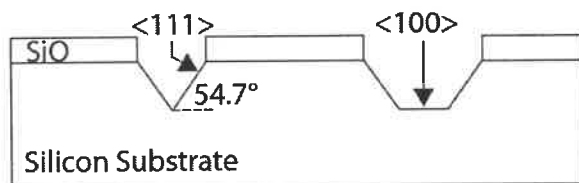


FIGURE 2. The KOH anisotropic etching profiles of (100) silicon.

wafer. Using standard lithography techniques, the masking layer can be patterned to expose three regions on the wafer for the etching of the grooves. When the wafer is wet etched in KOH, the three grooves will be formed simultaneously. Through this method it is possible to form near-atomically-smooth surfaces on the $\langle 111 \rangle$ plane, ideal surface-roughness conditions for a kinematic clamp. However, because of the crystal structure dependency, the stability and stiffness of the kinematic clamp is compromised. Figure 3 shows how KOH-etched groove on a standard (100) silicon wafer are aligned perpendicular to one another and how two grooves must be collinear.

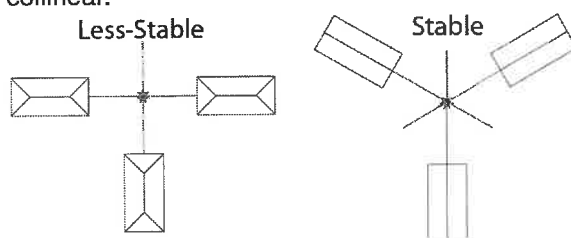


FIGURE 3. Crystallographic dependence of anisotropic etching affects the stability and stiffness of the kinematic clamp based on the geometry of the grooves.

A second method for groove fabrication makes use of Deep Reactive Ion Etching (DRIE). A DRIE process is capable of etching deep trenches with straight sidewalls into silicon wafers. Photolithography is used to specify the arrangement of the trenches. For a MEMS-kinematic clamp application, a masking layer would be deposited and patterned onto the wafer using standard lithography techniques. Using the DRIE process, deep rectangular trenches would then be etched into the silicon where the masking layer had been patterned. Using a chemical vapor deposition (CVD) process, these trenches would then be coated with a conformal film that would evenly round-out the edges to ensure point

contact between the groove and the ball-post, as shown in Figure 4. Depending on the application, the dimensions of the trenches are determined by the photolithographic mask and by the time-controlled deposition thickness of the CVD process.

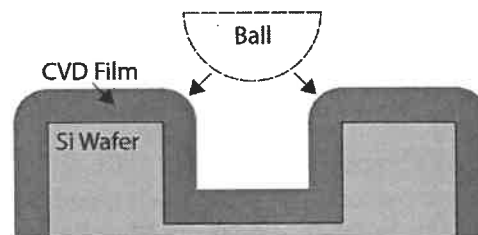


FIGURE 4. The CVD film rounds out the sharp 90° edges of the trenches and ensures point contact with the ball-posts. The relative amount of curvature depends on the aspect ratio of the trench and the thickness of the deposited layer.

This fabrication process for making "grooves" possesses a great benefit over the KOH wet-etched grooves: it does not rely upon the crystallographic orientation of the wafers and thus enables the fabrication of stable kinematic clamp configurations as shown on the right-hand side of Figure 3.

Ball-Post Fabrication

The fabrication of ball-posts was also investigated. Hemispherical voids are easily created in silicon wafers by isotropic wet-etching with agitation. These voids could be converted into positive features through a micro-mold making process with polymers; however, subsequent alignment steps required for this micro-mold making would sacrifice precision; furthermore, depending on the application, polymer materials may not be compatible with the devices.

An alternative method which resembles the DRIE groove etching process is the fabrication of ball-posts. Flat topped protrusions are easily fabricated using either isotropic wet-etching or DRIE processes by etching back the wafer and leaving "mesa-like" pillars on the surface of the wafer. These protrusions can be coated with a conformal CVD film in order to round out the edges to form near-hemispherical posts, as shown in Figure 5. Depending on the specific application, the dimensions of the initial post and the thickness of the CVD film must be tuned to match the groove. One consideration for the ball-post dimensions is mechanical failure from shear and buckling, be-

cause silicon is relatively brittle. To analyze the contact stress and stiffness, γ , the ratio of the ball-post radius of curvature to the groove's radius of curvature should be kept large if contamination at the interface is to be avoided; however, to minimize the effect of contact stress and deflection γ should be kept small[3].

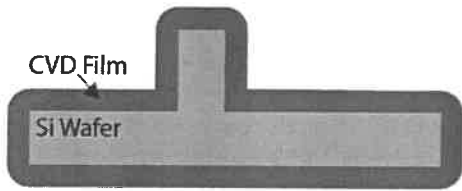


FIGURE 5. Conformally coated pillars generate hemispherical posts which will mate with the grooves.

Of the methods described above, the two best suited for fabricating a stable clamp are the DRIE etched grooves and the DRIE formed posts which are then both coated with a conformal CVD layer. The benefits of these methods are that the CVD layer is self-aligned to the groove and post features so only one optical lithographic step is required for each wafer. Additionally, because the processing steps for making the ball-posts are the same as those for making the conformally coated grooves the wafers can be processed side-by-side, saving time and reducing fabrication costs.

ERROR ANALYSIS

To determine the theoretical precision of the kinematic clamp, the fabrication tolerances of the tools in the UC Berkeley Microfabrication Laboratory were analyzed to determine the accuracy and precision of the fabrication process. Two methods of lithography were considered: Karl-Suss contact-mode mask lithography, with a positional tolerance of $2\mu m$ and a mask feature tolerance of $0.25\mu m$, and Crestec electron-beam lithography, which does not require a mask, and is capable of $10nm$ tolerances.

Modern wafer surface finishing techniques, such as chemical mechanical polishing, are capable of reducing surfaces roughness to as low as $2-3\text{\AA}$. Additionally, the SVG photoresist developer tool in the UC Berkeley Microfabrication Lab on has a thickness uniformity of 5% across the wafer, (a $5nm$ variation over a $100nm$ -thick film). Because of varying applications and varying feature sizes, the theoretical precision of the clamp varies and a specific example must be considered in order to

quantify the precision.

CVD processes are widely used in both research environments and industry. In industry, near perfect conformality is achievable along with nanometer accuracy in both thickness and uniformity. However, research labs do not achieve these precisions. In the UC Berkeley micro-fabrication laboratory, typical errors associated with CVD depositions are 2% uniformity across a wafer, and about $3\text{\AA}/\text{minute}$ variation in deposition rate repeatability [5]. For most semiconductor substrates, CVD deposited layers exhibit surface roughness is typically dominated by formation mechanisms present during the deposition such as nucleation, and coalescence of clusters [6]. Due to the vast variety of processes possible, further investigation of these effects on variation caused at sharp edges and the resulting variation in curvature is recommended once a specific process is identified.

For simplicity, a (100) silicon wafer, $100mm$ in diameter, is analyzed. The ideal positions of the post and groove interactions will be $3cm$ away from the center of the wafer at 120° spacings. If silicon carbide, a high wear-resistant material, is used for the chemical vapor deposition, we would expect a surface roughness of $70nm$ [7]. It is assumed that the wafer is free of devices before clamp fabrication, so that conformality of the photoresist will be optimal. Depending on the type of MEMS application, the grooves can be fabricated before the devices so that the wafer is blank at the start of processing. To analyze the theoretical worst-case scenario, the errors from the fabrication process modules are added together and by using geometry the offsets are determined, as illustrated in Figure 6.

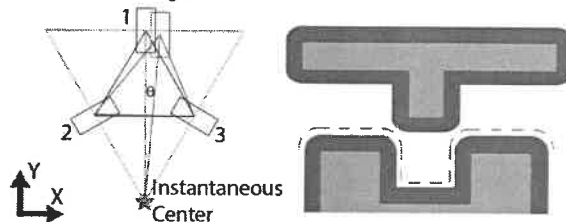


FIGURE 6. Visualization of rotational error (left), and the vertical positional error (right) resulting from fabrication tolerances.

For this theoretical example, if the impact of mis-positioning one groove is analyzed, it is determined that the fabrication tolerances produce a

worst-case rotational error of $2.037 \times 10^{-4}^\circ$ about the z-axis when using the Karl Suss contact-mode lithography tool, and $5.09 \times 10^{-5}^\circ$ about the z-axis when using electron-beam lithography. Because one mask is used in the optical lithography process and accurate transfer from the mask to the photoresist is assumed, this error mostly stems from the tolerances of the mask. For electron-beam lithography, tighter tolerances are made possible, and so the theoretical precision of the clamp is improved. Vertical position errors stemming from the CVD thickness and CVD surface roughness would be the same for all three sets of ball-post and groove interactions, so the vertical misalignment would be a function of the ball-post's radius of curvature.

CONCLUSION

The goal of the MEMS-scale Kinematic Clamp is to achieve six points of contact between ball-posts and grooves in order to deterministically constrain two silicon wafers kinematically. Methods for ball-post and groove fabrication have been analyzed and compared. The most stable and feasible method for the fabrication of both the grooves and the ball-posts is accomplished by the deep reactive ion etching of the silicon wafer, followed by a conformal chemical vapor deposition to round the edges. In the case of the grooves, trenches are etched out from the wafer, while for fabricating ball-posts, the majority of the wafer is etched back, leaving behind posts. Either electron-beam or optical mask-contact lithography can be used for patterning the positions of the clamp's ball-posts and grooves; however, higher theoretical precision is achieved by using electron-beam lithography. The exact dimensions of the clamp are specific to the application of the clamp, and a thorough analysis of the tolerances encountered during fabrication must be conducted.

Depending on the geometry of the final design, it is possible that both the top and bottom of the clamp could be etched together and put into a CVD furnace together. The only difference would be the mask layout. In addition, both of these processes lend themselves well to batch processing, which is a requirement for inexpensive MEMS production. The clamp's ball-posts and grooves must be tailored for each specific MEMS applications and this analysis would need to be performed for those dimensions to determine if a kinematic clamp is a feasible solution for a given design.

FUTURE WORK

Continuation of this work would consist of fabricating the MEMS-scale kinematic clamp using the methods described in this paper. In order to verify the calculations, a test rig would be developed and the clamp's repeatability and precision could be tested with a coordinate measuring machine, which is readily available at the Lawrence Berkeley Laboratory. Additionally, failure criteria of the clamp could be investigated to determine what geometries would best suit certain loading conditions. If commercially viable, the kinematic clamp could be implemented in the mainstream MEMS market for various applications.

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ULTRA-PLANAR ELECTROSTATIC CHUCKS BASED ON LOW CTE MATERIALS FOR LITHOGRAPHY AND METROLOGY

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MOTIVATION

In the race for more computing power and shorter pathway in micro electronic circuits, the limits are set by the lithographic structuring process. Extreme Ultraviolet Lithography (EUVL) and various Electron-Beam Direct-Write techniques (e-beam) are promising techniques for future production at lower nodes. A reliable positioning of silicon wafers and masks in a high vacuum environment are required and the achievable substrate flatness strongly influences the outcome. Ultra-planar electrostatic chucks offer the possibility of holding and leveling substrates in a high vacuum. Precision mechanics based on low CTE (coefficient of thermal expansion) materials like glass or glass ceramics combined with optical manufacturing techniques ensure high exposure qualities. The application of chucks is not limited to lithography. Similar needs exist in many measurement techniques.

INTRODUCTION

So far, mechanical actuators or vacuum grippers are used to keep wafers under ambient conditions. Mechanical clamps are widely used under vacuum conditions. The residual wafer bending can be calculated. But mechanical clamps generate high local loads at the point of contact and therefore particles and deformations.

Electrostatic chucking is an alternative to mechanical clamps. An electrostatic force is generated by an electric field between the substrate and the chuck electrode. This clamping force is distributed homogeneously over the whole surface and can be easily adjusted via the applied voltage. By using an electrostatic chuck, flattening of bended wafers or masks is possible, if forces are sufficiently high and the chuck itself is smooth and stiff.

This article presents the design, manufacture and testing of various electrostatic chucks as a central component of ultra-precision components for lithography and metrology tools.

BASICS OF ELECTROSTATIC CHUCKING

The basic design of an electrostatic chuck is similar to a parallel plate capacitor. It is possible to design a unipolar (one electrode) or bipolar (more than one electrode) configuration. The substrate has to be electrically conductive (intrinsic conductivity of Silicon is sufficient) or must be coated with a thin metallic layer.

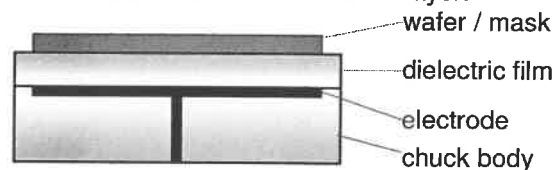


FIGURE 1. Schema of an electrostatic chuck

Unipolar design

When the substrate is placed on the chuck and a voltage U is applied between the two plates, Coulomb forces build up and attract the substrate to the chuck with a force F proportional to the electric field. Assuming an ideal dielectric film, the force F per area A can be calculated by

$$F_{\text{uni}} = \frac{1}{2} \epsilon_0 \epsilon^2 A (U/d)^2, \quad 1$$

where d is the distance between the two plates and ϵ the relative electrical permittivity of the dielectric [1]. An electrostatic pressure of about 10 kPa may be obtained with a voltage of about 1 kV and a perfect dielectric with a thickness of about 100 μm . Two principles of chuck designs are shown in FIGURE 2.

Bipolar design

The bipolar design ensures clamping for all types of semiconducting wafer and metal coated mask without the need of an electrical contact to the substrate. By using suitable electrode geometry and wiring of the electrodes, a floating potential is generated on the substrate.

Typically, two equally sized chuck electrodes are loaded with inverse polarity. Therewith, the substrate electrical potential is close to zero.

Essentially the same electrostatic pressure value as with a unipolar chuck can be obtained. The schema is shown in FIGURE 2.

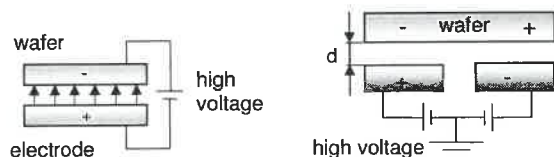


FIGURE 2. Principle of chucking, unipolar (left) and bipolar (right)

MATERIALS

Mechanical, thermal and vacuum-technical demands influence the choice of material. Generally, a high bending stiffness D is essential. The bending stiffness is sensitive with respect to the material thickness h , the modulus of elasticity E and the Poisson's ratio ν .

$$D = E h^3 / 12 (1 - \nu^2) \quad 2$$

The thermal behaviour of the assembly is determined by the CTE of the material. Special materials with low thermal expansion are necessary to achieve stability within a range of a few nanometres, see TABLE 1. Particularly, the glass ULE by CORNING and the glass ceramics ZERODUR by SCHOTT are applied. Both materials have a thermal expansion in the range of $1.2 \times 10^{-8} \text{ K}^{-1}$ and an excellent stability in high vacuum atmosphere (notably low out-gassing).

TABLE 1. Material properties

Parameter	SiO ₂ glass	ULE glass	ZERODUR glass ceramic
Young's Modul [GPa]	72	67.6	90.3
Poisson's ratio		0.17	0.243
Density [g/cm ³]	2.2	2.21	2.53
CTE [K ⁻¹] (roomtemp.)	5×10^{-7}	1×10^{-8}	2×10^{-8}
Thermal conductivity [W/m K]	1.31	1.31	1.46
Dielectric constante (100 Hz)	3.8	4	8
Typ of dielectric	Coulomb	Coulomb	Johnson- Rahbek

The type of the dielectric material is essential for chucking. With a so called Coulomb dielectric, a nearly perfect isolation exists. The clamping force is constant in time. The behaviour of a so called Johnson-Rahbek dielectric is totally different because the dielectric is virtually "semiconducting". Charge enters the dielectric material and builds an electric double layer close to the substrate interface [2].

This leads to clamping forces up to 5...10 times higher than with Coulomb systems. However, the clamping force is time dependent. The substrate may adhere some hours after switching off the power.

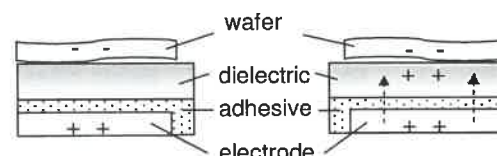


FIGURE 3. Coulomb dielectric (left) in comparison to Johnson-Rahbek dielectric (right)

DESIGN

A chuck is composed of a body and a dielectric FIGURE 1. For the body, often ZERODUR is chosen because of its high elastic modulus. The thickness is optimised according to the bending stiffness. The dielectric film is generally made of ULE. Both components are connected by adhesive. The typical CTE of epoxy adhesives varies between $20 \dots 60 \times 10^{-6} \text{ K}^{-1}$ and has to be considered in the low CTE design. The long time stability of adhesive is also restricted but there where no limitation detected in many years of operation. The dielectric constant of adhesive also influences the clamping force.

The chuck is fixed by a 3-point mount using V-grooves and spheres. The mount points are arranged in the optimal position of minimal gravity effect and minimize mechanical stress [3].

The mass budget for moving components is a very important point. A light weight solution was developed with a stiffness nearly equal to the solid body but with less than 50 % of its mass. Holes run parallel to the closed back side, arranged in orthogonal crosses form self-supporting vaults. The distance between chuck surface and light weight structure was optimized so that no effects of holes are transferred to the surface. The diameter of the holes (10 mm...12 mm) was designed in relation to the thickness of the body. It is possible to arrange holes in two different levels. The space between

two holes must not be lower than 3 mm; a typical value is 5 mm. To avoid micro cracks and to increase the breaking strength all surfaces have to be etched finally. A value of 0.1 mm to 0.2 mm for the etch depth is expedient.

The surface of the dielectric is structured with a pin pattern with a height of about a few micrometers and a pitch of a few millimetres. The pin pattern is optimised for maximum chucking force and minimal deformation of the substrate. A guideline for Si-wafers is: pin diameter: 1 mm; pin height: 3 µm; pin pitch: 3 mm. Besides, pin pattern are useful for the de-chucking process and – even more important – for reducing the risk of flatness distortions by particle contamination at the interface chuck to substrate. As a matter of fact, particle contamination is inevitable during manufacturing. With this knowledge, the area of the structured surface was reduced to less than 5%.

MANUFACTURING AND CHARACTERISATION

At IOF, a technology analysis has started to identify suitable manufacturing processes for chucks made from low CTE materials. The result is a combination of optical technologies with micro systems technologies. The chuck body and the dielectric will be manufactured by classical optical polishing processes. The achieved flatness is better than 1 µm (p.-v.). The assembly of both components is realized by a vacuum compatible epoxy with low viscosity and low out-gassing ratio. The challenge is to achieve a thin polymer film with a thickness of less than 10 µm.

The bulk dielectric is machined to a thickness of lower than 200 µm after the bonding process. In an iterative chemical-mechanical polishing process based on a one disc polishing machine, the final thickness and flatness are achieved. This part of manufacturing is done in cooperation with Jenoptik Laser Optic Systems Jena (Germany).

The manufacturing steps are controlled by interferometer measurements. A flatness typically less than 200 nm (p.-v.) over an area up to 12 inch is achieved. Optical manufacturing technologies work well on a circular plate with a closed surface and much higher flatness values can be obtained than with a rectangular shape or with openings / holes in the surface.

For process control and the characterisation of the chuck surface, a Fizeau interferometer, type

12 inch ZYGO GPI VeriFireAT with a high resolution camera, is used. The reference transmission flat has a quality of 15 nm (p.-v.). A model of measuring setup is shown in FIGURE 5. The interferometer measures in horizontal direction but stepper uses the chuck vertically. The resulting gravitational effect can be simulated and considered mathematically.

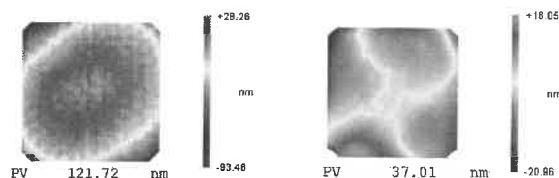


FIGURE 4. Interferogram before and after chemical-mechanical polishing process

Afterwards, the surface is structured in a lithography process. The pin pattern is manufactured by a plasma-supported dry etching process. Nearly every pattern can be structured. To cover the areas that should not be etched, a metallic layer is used. The etching ratio of the layer is not as high as the etching ratio of the glass. The realised pin structures are very homogeneous in height and have smooth edges.

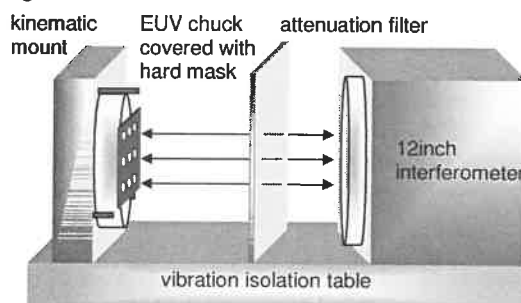


FIGURE 5. Measurement setup

APPLICATIONS

EUV-Mask Chuck

For next generation lithography (EUVL) the chuck functional parameters and suitable materials are specified in a recent standard proposed by SEMATECH [4]. In particular, the chuck flatness is highly challenging, because the deviations are limited to < 6 nm for a square area of (20 mm)² and to < 50 nm for the whole mask size of (142 mm)². To ensure the flattening of a standard EUV mask made from 6.35 mm thick “zero-expansion” glass, a chuck of high bending stiffness > 30 kNm and a

clamping pressure of $15 \text{ kPa} \pm 10 \%$ is demanded.

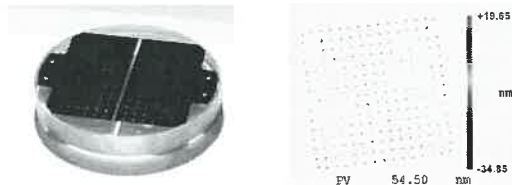


FIGURE 6. EUV mask chuck and interferogram

FIGURE 6 shows the chuck prototype (based on the results of the preliminary model [5]). The chuck height is 28 mm. The bending stiffness is calculated to about 176 kNm. The materials used for the chuck dielectric as well as for the chuck body are transparent; so the metallic chuck electrodes are clearly visible as rectangular areas in FIGURE 6. Two electrodes are used in a symmetric bipolar configuration. There is a rectangular pattern of pins within the quadratic chucking area. The demonstrated flatness was 54.5 nm in the quality area.

E-beam Chuck

For applications in direct write e-beam lithography large electrostatic wafer chucks are required. FIGURE 7 shows a 12 inch diameter wafer chuck developed and manufactured by IOF from "zero-expansion" materials.

There are three holes visible at about half the wafer radius. These allow a lifting mechanism to transfer the wafer to a handling unit. The flatness of the non structured chuck surface is in the range of less than 200 nm across the whole diameter and even an order of magnitude better in any local field of 20 mm x 20 mm. A closer look to FIGURE 7 shows, that there is a fine patterning on the chuck surface, i.e. the wafer supporting area is again realised by a large number of small pins.

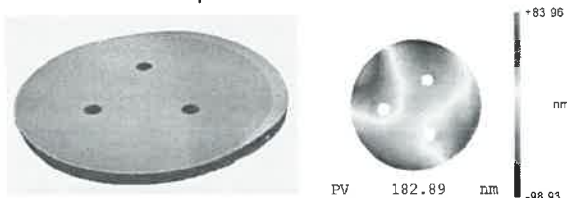


FIGURE 7. 12" wafer chuck for e-beam application

Chuck for Metrology

X-ray fluorescence spectroscopy using Synchrotron light sources, is a powerful method for the characterization of wafer materials.

Again, a reliable positioning of Si-wafers in a high vacuum environment is required.

For a synchrotron application, a compact 12 inch electrostatic chuck with an integrated lift mechanism for smooth wafer handling onto and off the chuck was developed and manufactured, see FIGURE 8. Wafer substrates can be clamped for both horizontal and vertical orientations. The chuck works bipolar and the flatness is approximately $2 \mu\text{m}$ p.-v. Electrical tests verified that a voltage of less than 500 V was sufficient for safe clamping of Si-wafers vertically.



FIGURE 8. 12" Wafer Chuck for spectroscopy

CONCLUSION

Electrostatic chucks using "zero expansion" materials are well suited for clamping wafer and mask substrates for lithography and metrology applications in vacuum. For next generation lithography, ultra-planar chucks with flatness of nearly 50 nm have been obtained and even better values are realistic in the near future.

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COMPACT, MULTI-CHANNEL, FIBER-BASED ABSOLUTE DISTANCE MEASURING INTERFEROMETER SYSTEM

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INTRODUCTION

This paper describes the principle of operation, architecture and performance of a high-accuracy displacement measurement sensor with absolute distance measurement capability [1]. The system is optical fiber-based and leverages the reliability, flexibility and availability of telecom components. While other fiber-based systems have been implemented [2], this particular system is designed for demanding applications requiring a multitude (~60) of thermally passive, electrically immune, compact sensors with an extremely long mean time between failure (MTBF) and exceptional long-term drift performance.

PRINCIPLE OF OPERATION

This section describes the principle of operation of the two operating modes of the sensor system.

Displacement measurement

Displacement measurement refers to a relative measurement that tracks the displacement of a target from some arbitrary origin and provides no information about the *distance* of the target from the sensor. This measurement mode is based on a heterodyne coupled cavity arrangement shown schematically in Figure 1.

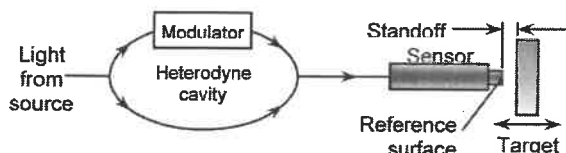


Figure 1. Schematic of coupled cavity

Light from a relatively broadband source enters the first cavity which is called the heterodyne cavity. The incoming light is split into two parts of slightly different path lengths, the difference being related to the sensor standoff. One half passes through a modulator that imposes a frequency shift on that portion. The light from the two paths then recombines at the exit of the heterodyne cavity. The frequency shifted and unshifted beams then enter a second cavity

which is comprised of the reference surface of the sensor and the reflective target. Portions of both beams are reflected from the reference surface and the target. The components that are reflected from the target acquire an additional optical path length due to the round trip to the target. This additional optical path length matches the path length mismatch in the heterodyne cavity and interference can now occur between one of the beams reflected at the reference surface and a beam of different frequency reflected from the target. The resulting AC interference signal (which is at the frequency imposed by the heterodyne cavity) can now be processed using phase measuring techniques. This optical configuration confers several important advantages:

- All active heat generating components are separated from the passive sensors
- Sensor standoff and measurement range are independent of one another
- Measurement and reference beams are collinear eliminating any Abbe offsets

Absolute distance measurement

Absolute distance measurement refers to the determination of the absolute distance of the target from the sensor reference surface. The ability to measure the absolute distance provides a "homing" functionality, i.e., it allows for repeatable repositioning of a target mirror at a predetermined distance from the reference surface of the sensor. This functionality is implemented using three-wavelength interferometry and the method of exact fractions due to Michelson and Benoît [4]. In this method the fringe order is obtained through the use of a "synthetic wavelength" of a length appropriate to the maximum distance of the target from the reference surface. The synthetic wavelength is derived from two wavelengths that are selected to produce the desired synthetic wavelength. The fringe fraction information obtained from the synthetic wavelength is used to resolve fringe

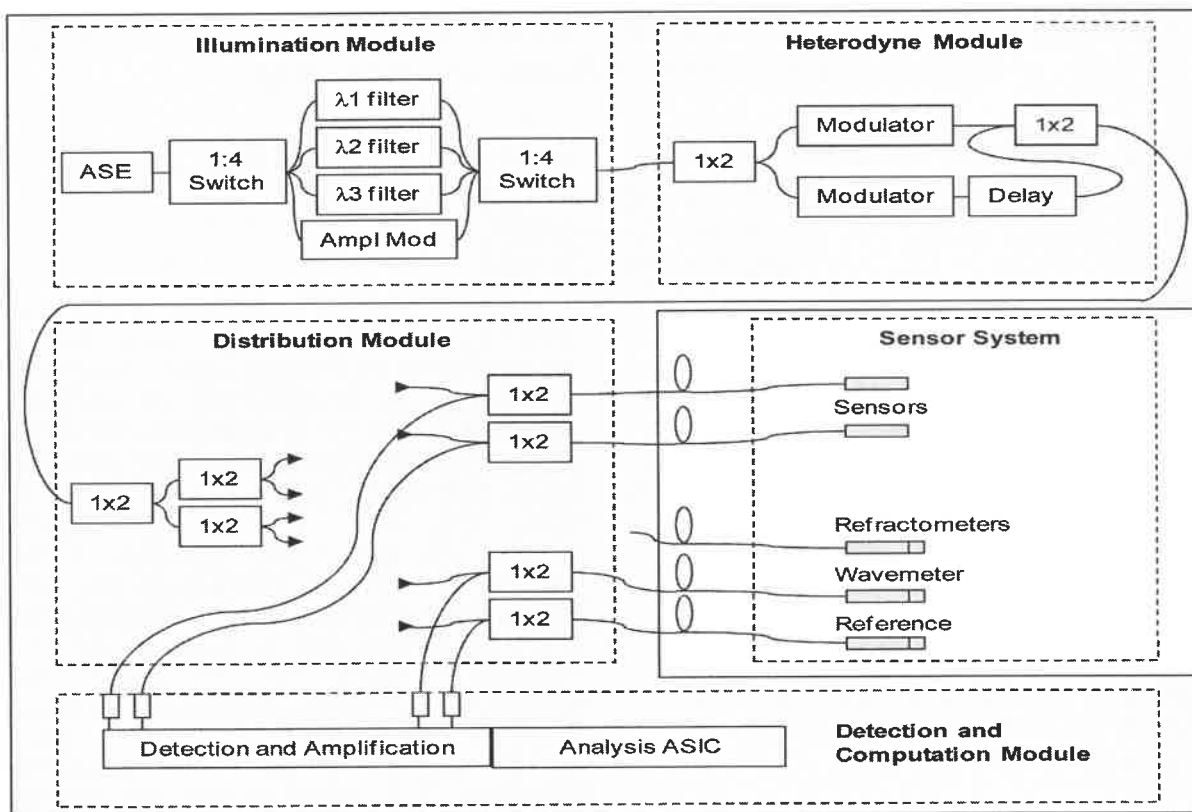


FIGURE 2. System block diagram showing major components. Shaded region indicates Control Unit.

order ambiguities in one of the shorter wavelengths and combined with the fringe fraction information at this wavelength to refine the estimate of the position of the target.

SYSTEM ARCHITECTURE

The system can be divided into five functional units as shown in Figure 2; an illumination module, a heterodyne module, a distribution module, a detection/computation module and the sensing subsystem. A typical measurement sequence proceeds as follows: The light source is sequentially cycled through the three wavelengths in order to establish the absolute distance of the target for each channel. The position of the target is then tracked from this initial position using a single wavelength in relative measurement mode.

Control Unit

The control unit comprises the first four modules and parts of the sensor subsystem. The sensor subsystem is connected to the control unit via optical fibers. The system is designed such that items that may require maintenance or replacement are confined to the control unit while the high-reliability sensors are embedded

within the system being monitored. This partitioning is a feature of the design that ensures that any maintenance or replacement does not require disassembly of the system being monitored.

Illumination module (IM)

The IM provides illumination at three selectable wavelengths. This is achieved by means of a broadband amplifies spontaneous emission (ASE) source, the spectral output of which spans the desired wavelengths. The three individual wavelengths are produced by filtering the output of the ASE using custom transmission filters centered at the three base wavelengths (1530 nm, 1530.75 nm and 1560 nm). MEMS switches capable of rapid switching are used to select the desired filter. This choice of wavelengths produces synthetic wavelengths of 80 μm and 2 mm for use in the absolute distance mode. A fourth option sends the raw output of the ASE through a modulator for the purpose of performing system calibrations.

The spectral shape of the filter output is critical to the system performance as it determines the coherence length (and consequently the measurement range) and also the linearity of the

sensor output. Linear performance requires symmetric spectral shape that drops to zero smoothly at the edges of the passband. The filters are designed to produce a Gaussian shaped spectrum with a FWHM of ~ 3 nm.

Heterodyne Module (HM)

The light from the IM at a given wavelength enters the HM where it is sent through an unbalanced Mach-Zehnder interferometer where at least one path is phase shift modulated using serrodyne modulation [5]. In this technique, the phase is modulated using a sawtooth drive signal which is adjusted to produce a phase change of 2π radians. The flyback time is minimized by the use of a high-speed modulator, with the net result that the serrodyne simulates a continuously varying linear phase change of one path relative to the other. The AC signal that results from the interference of the phase-shifted and unshifted components is used in subsequent measurements of phase. The imbalance in the two path lengths is an independent parameter that is determined by the sensor standoff.

Distribution module (DM)

The DM is a passive module that employs a cascade of 1X2 splitters to distribute the output of the HM into 64 measurement channels and to extract the return signal from each channel and route it to the detection module.

Detection and computation module (DCM)

The DCM converts the signal from each sensor into a phase/displacement measurement. The phase measurements from the reference and wavemeter are used to compute the source wavelength. This information is combined with the refractive index measurements of the refractometer and used to generate a scale factor which is used to convert the measured phase from each sensor into a displacement or distance.

Sensor subsystem

The sensors used in this system are variations of a Fizeau interferometer geometry and are used to measure position and refractive index. The various sensor types are adaptations of the basic straight-through sensor that is shown schematically in Figure 3. As shown in the figure, the sensor consists of a fiber whose end face is separated from a Gradient Index (GRIN) lens. The last surface of the GRIN lens acts as the reference surface for the Fizeau cavity. All

measurements are made relative to this surface and this surface is accessible to serve as datum surface for mounting purposes.

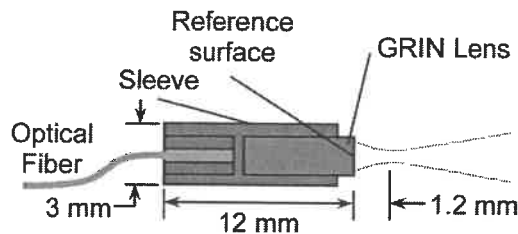


Figure 3. Schematic of sensor.

The beam waist of the exiting beam is set to a fixed working distance from the end of the GRIN. These basic sensors are extremely compact, stable and inexpensive.

This basic design is used for position sensing by placing a reflecting target at the beam waist position. Interference between the wavefronts reflected from the GRIN reference surface and the target produces the signal used to measure the position of the target.

This basic design is used in conjunction with a fixed cavity (etalon) in the construction of the reference, wavemeter and refractometer sensors. The reference sensor measures the interference signal from the etalon and is used to measure and compensate the drift of the HM. In this application, the reference surface of the GRIN lens is antireflection (AR) coated as the signal of interest is the interference generated between reflections from the two surfaces of the etalon. The wavemeter also incorporates a similar etalon of a slightly different length and in conjunction with the reference sensor is used to make a continuous measurement of the wavelength. The refractometer measures the absolute refractive index of the medium in which the sensors are immersed. It is identical to the reference etalon with the difference that the etalon cavity is open to the medium of interest. The absolute length of the etalons used in the various monitors must be known with low uncertainty and is determined by a technique that combines Fourier transform interferometry [6] using a widely tunable laser source and angle scanning interferometry [7]. The former technique determines the optical path length of the etalon with an uncertainty of ~ 10 nm. This information is fed to the latter technique which makes a determination with an uncertainty of ~ 1 nm [7].

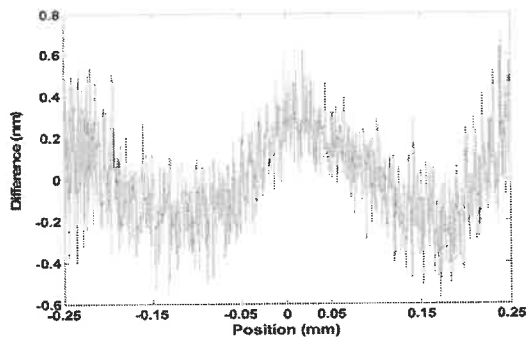


Figure 4. Displacement measurement performance.

DISPLACEMENT PERFORMANCE

The displacement measurement performance of the system was assessed by comparing the results to simultaneous measurements of the displacement of a target by a commercial HeNe based displacement interferometer (Zygo ZMI-4000). A detailed description of the validation metrology setup used to make this comparison may be found elsewhere in these proceedings. Only the results are summarized here. The difference between the displacement measured using a single wavelength by the system and validation metrology is shown in Figure 4 as a function of target displacement. The difference shown is the residual after removal of a ~ 10 ppm linear term. This linear term appears to result from an error in the measurement of the wavelength in this preproduction units.

The residual that is observed should be viewed in the context of the uncertainty in the measurand, which in this case is the difference. The $k=2$ expanded uncertainty of the difference parameter is estimated to be ~ 0.5 nm. This suggests that measurements agree to within the uncertainty in the difference, i.e., within 1 ppm.

ABSOLUTE DISTANCE MEAS. DRIFT

Drift performance is assessed by repeated measurements of a stable fused silica air-gap etalon of 1.4 mm length and is shown in Figure 5. This result displays the stability of the absolute distance measurements and are an indication of the stability of a home position established using the sensor. The etalon was measured in the absolute measurement mode every 40 seconds over a period of 14 hours and shows a PV drift of ~ 2.5 nm, suggesting that the sensor can provide homing capability at the level of a few nm.

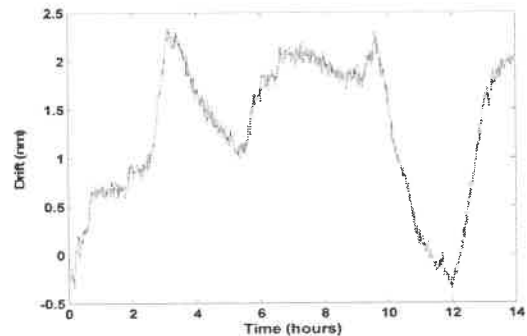


Figure 5. Drift in absolute measurement

SUMMARY

A new compact, multi-channel, fiber-based Interferometer system capable of both displacement measurement and absolute distance measurement has been described. Data that show the level of performance currently attainable in each of these modalities has also been presented.

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A STUDY OF MODULATED INTERFERENCE LASER ENCODER NEW APPROACH FOR A HIGH RESOLUTION AND HIGH STABILITY POSITION MEASUREMENT SYSTEM

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BACKGROUND

Industries in nanotechnology, mass data storage, semiconductor equipment require highly-reliable machines with extremely fast and stable positioning capability at sub-nanometer level. Today, laser interferometers, capacitance gauges and interference laser encoders have achieved sub-nanometer resolution; however, each comes with their own issues. Laser interferometer operated in air is sensitive to minute changes in air temperature, humidity and atmospheric pressure, which hinders repeatable measurement. Capacitance gauge is often limited by its short measurement range and difficult alignment process during the initial set-up.

As a result, the demand for position encoders has been increasing despite the fact that a high-end position encoder often requires meticulous installation and signal adjustment procedure due to the extremely small pitch grating (~0.5 micron) and short working distance employed in its basic design. Furthermore, conventional position encoder may produce unacceptable interpolation errors without proper signal adjustments.

CONCEPT OF MODULATED ENCORDER

SCAN MODULATED OPTICS

Figure 1 illustrates the configuration of the new scanner modulated encoder which integrates an optical scanning mirror and conventional interference encoder optics using a transmission type scale. The reflection type scale can be also used with similar optical set-up.

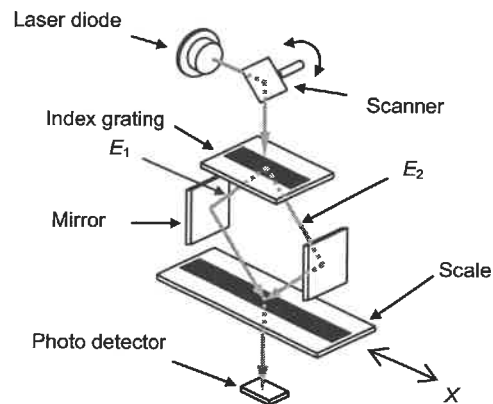


FIGURE 1. Concept of modulated interference laser encoder

Coherent laser beam is angularly modulated by the scanning mirror and enters through the index grating, which divides the beam into +1/-1 order diffracted lights. Each diffractive light are bent by mirrors and incident to moving scale surface. This scale is of the same pitch grating as the index. Lights are diffracted again in same direction. The +1/-1 order lights can be represented as Equations (1) and (2) by complex amplitude expression.

$$E_1 = A_1 \exp\left(jkL_1 + \frac{2\pi X}{P}\right) \quad (1)$$

$$E_2 = A_2 \exp\left(jkL_2 - \frac{2\pi X}{P}\right) \quad (2)$$

, where A_1 , A_2 are amplitudes of the light intensity and L_1 , L_2 are nominal optical path lengths respectively. P is the pitch of the moving

scale and the index grating. X is the relative position between the index grating and the moving scale. Assuming that the incident angle and thus the diffraction angle of the light entering into the index grating is modulated in a single frequency sinusoidal waveform, the complex amplitude of +1/-1 order diffracted lights can be described as;

$$E_1 = A_1 \exp \left\{ jk(L_1 + M \sin \omega t) + \frac{2\pi X}{P} \right\} \quad (3)$$

$$E_2 = A_2 \exp \left\{ jk(L_2 - M \sin \omega t) - \frac{2\pi X}{P} \right\} \quad (4)$$

, where M is the amplitude of the modulation, and ω is the angular frequency. Assuming that $L_1=L_2$, interference intensity distribution I_1 is then calculated as:

$$I_1 = |E_1 + E_2|^2 \quad (5)$$

$$= A_1^2 + A_2^2 + 2A_1A_2 \cos\left(\frac{4\pi X}{P} + 2M \sin \omega t\right)$$

Figure 2 illustrates the signal form of I_1 . The interference power I_1 wave-form changes as the relative position between the index grating and scale changes from $X=P1$, $P2$ to $P4$.

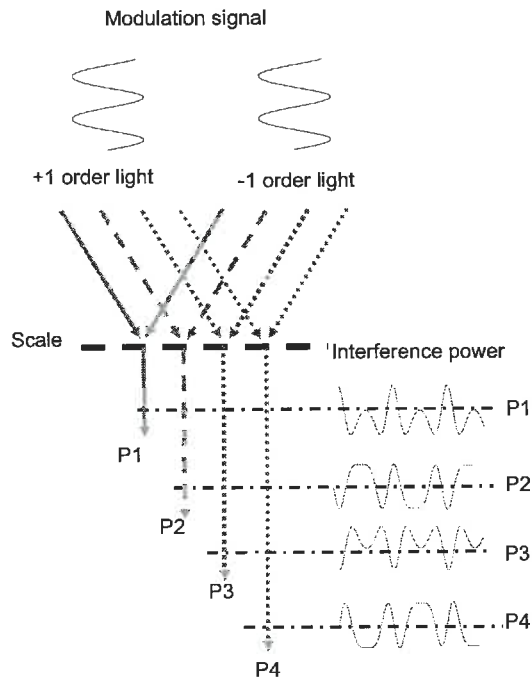


FIGURE 2. Modulated interference signal

POSITION DECODING LOGIC

Equation (5) shows that I_1 is now a phase modulated signal, which is a function of time. The Fourier expansion analysis of Equation (6) in time domain shows:

$$I_1(t) = A_1^2 + A_2^2 + 2A_1A_2J_0(2M) \cos\left(\frac{4\pi X}{P}\right) \quad (6)$$

$$- 4A_1A_2 \sin\left(\frac{4\pi X}{P}\right) \sum_{m=1}^{\infty} J_{2m-1}(2M) \sin\{(2m-1)\omega t\}$$

$$+ 4A_1A_2 \cos\left(\frac{4\pi X}{P}\right) \sum_{m=1}^{\infty} J_{2m}(2M) \cos\{2m\omega t\}$$

Where the function $J_n(2M)$ represents Bessel's Function of the first kind. This result reveals that the scale position information is expressed as amplitudes of the high order harmonics of the scanning frequency.

Although there can be many methods to decode the position information from Equation (6), reference [3] describes the phase detection method using an all-digital PLL (Phase-Locked Loop), which first multiplies the incoming signal described in Equation (6) by $\cos(0.5\omega t)$ and then use $\sin(2.5\omega t)$ for extracting the position information, X in Equation (6).

Figure 3 is the block diagram of the detection logic SPPE developed by NanoWave Inc. There is a synthesizer within the logic, which generates three synchronized phase waves $\cos(0.5\omega t)$, $\cos(0.5\omega t)$, $\sin(2.5\omega t)$, one for the scanner in the optical head and two for the position decoder.

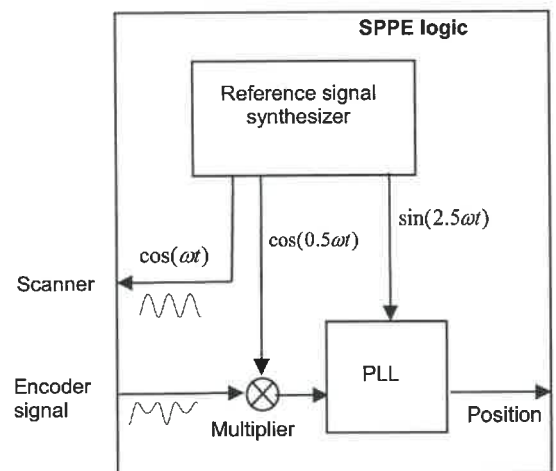


FIGURE 3. Position decoding logic

PRODUCTION DESIGN

OPTICS

Figure 4 illustrates the optical configuration of the encoder head used in the actual product model. Instead of using a transmission type scale described in Figure 1, the production model is designed to use a reflection type scale for ease of installation and space saving purposes. In order to avoid heat transmission from the laser diode, laser light is guided through a single mode optical fiber. Laser beam incident from the fiber edge is modulated by a tiny optical scanner and then collimated. The optical scanner is made from quartz crystal which has twisted mechanical resonance of around 200 kHz. The index grating consists of 4μm pitch grating. Figure 5 is a photo of the actual optical head.

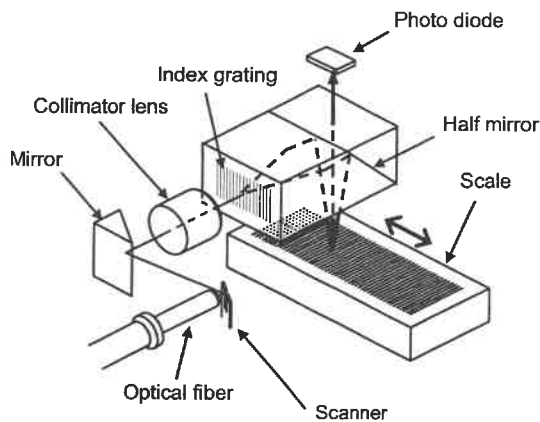


FIGURE 4. Production model optics

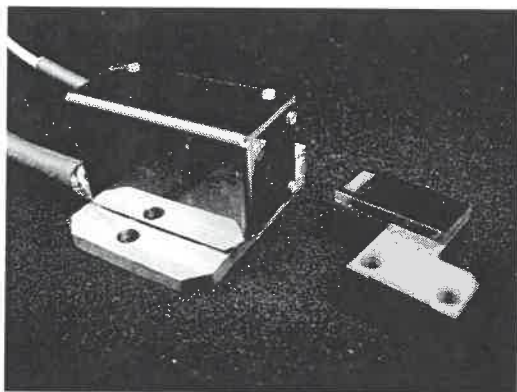


FIGURE 5. Optical head and scale

LOGICS

Figure 6 illustrates the system configuration of the scanning encoder. A ~200 kHz reference frequency signal is generated by the synthesizer inside the FPGA board and sent to the scanner through DAC, which drives the optical scanner. The FPGA logic continuously adjusts the reference frequency so that the scanner is always driven at near its peak resonant frequency. This way, the input energy to the head is minimized while the phase noise associated with the position detection stays low. All logic is implemented in a compact and low cost FPGA package.

Figure 7 shows the position information detected through the SPPE logic. The position data LSB 7.6pm and spectrum noise density is about 1pm/√Hz has been achieved.

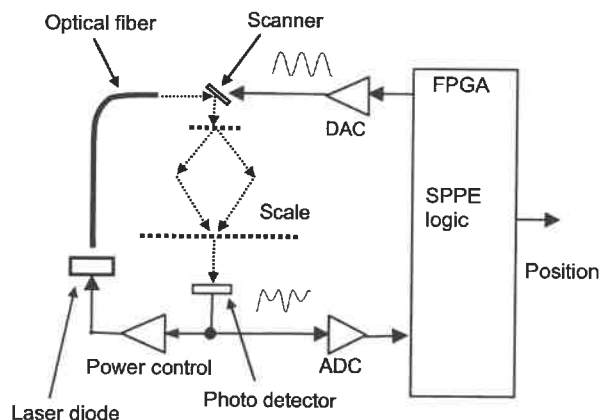


FIGURE 6. System configuration

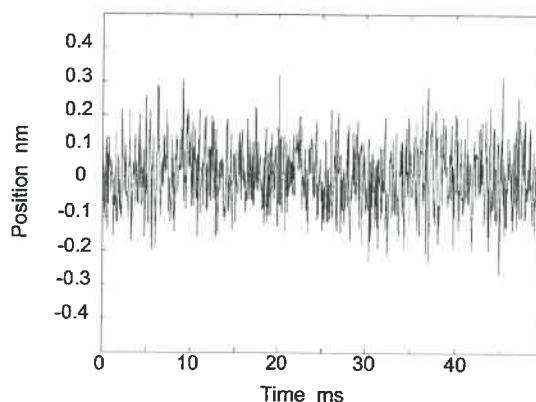


FIGURE 7. Position signal

EVALUATION

Figure 8 shows the result of the long term stability test. The optical head and the scale are fixed within the stainless steel block. Then, its position output is recorded for one month. The total drift of the position output is within 10 nm, which is comparable with high-end conventional encoders.

Figure 9 shows the angular error tolerance data. Large grating pitch ($4\mu\text{m}$) of this scanning encoder allows over $1.5'$ (0.44 mrad) tolerance.

CONCLUSIONS

A new concept position encoder product, which integrates interference encoder optics and the phase detection logic called SPPE (scan position probe encoder) was presented. The product design allows the use of 4 micron pitch grating while achieving noise spectrum density of $1\text{pm}/\sqrt{\text{Hz}}$ RMS with 7.6pm LSB resolution. Its relatively long grating pitch significantly simplifies the installation process and results in great robustness and tolerance in attaining the specified accuracy.

Furthermore, its optical system proved to be highly stable for long term measurement, achieving only a few nanometer fluctuations over a 1 month period test.

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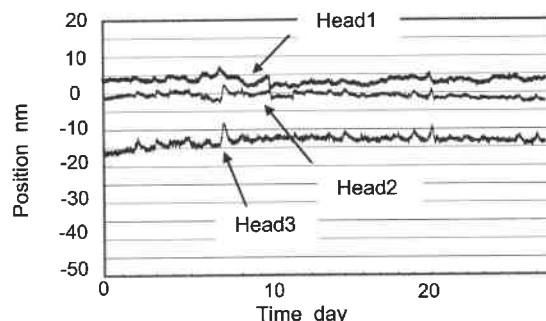


FIGURE 8. Long term stability

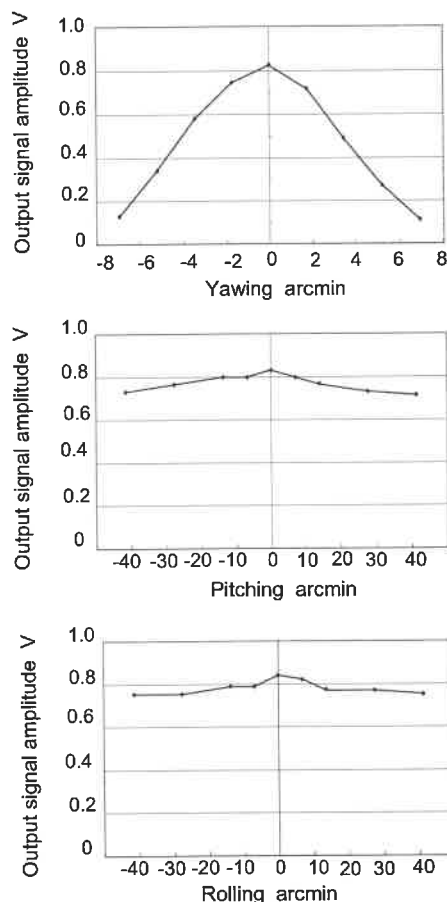


FIGURE 9. Mounting angle tolerance

INTERFEROMETRICAL SYSTEM FOR HIGH PRECISION MEASUREMENTS OF FLATNESS, THICKNESS AND PARALLELISM OF MECHANICAL PARTS

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INSTRUCTIONS

In the past, interferometry was a measurement method used mainly by the optical and semiconductor industries. Today there is an increasing amount of optical measurement systems used for applications in machine tools and the automotive industry which were previously performed primarily by tactile measurement systems.

There is an explanation for this transition in technology. With the ongoing development in the automotive industry to improve energy efficiency, as well as, to achieve lower emission values, there is a need for higher precision parts. The measurement techniques follow this manufacturing trend resulting in measurement systems with nanometer resolutions and capabilities. These systems have the ability to measure tolerances of less than $1\text{ }\mu\text{m}$ and with today's requirements this is very important.

Advanced development of optical measurement methods such as white light interferometry allow for the ability to build high precision measuring machines that can fulfil the accuracy and stability needs of large-scale production. However, the large range of various shaped parts could require special techniques of interferometric measurement applications.

MEASURING SYSTEM

The optical system shown in Fig.1 is based on a concept of conventional Michelson white light interferometry with a depth resolution of 1 nm .

One basic advantage of a white light interferometer is that the depth resolution is independent of the aperture of the objectives. The high depth resolution is necessary in order to perform form measurements of mechanical parts with tolerances of less than $1\text{ }\mu\text{m}$. Similar setups are also described in the literature [1], [2].

The reference mirror and the object are illuminated by a plane wave. Both the reference mirror and the object are projected onto the camera.

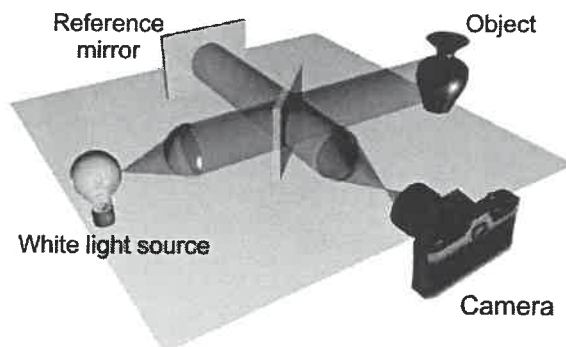


FIGURE 1. Optical scheme of an interferometeical head – white light interferometer.

White-light interference displays the maximum contrast when the optical path length in the reference arm is equal to the optical path length in the object arm. In order to obtain the 3D shape of the object, a change of the path lengths in the reference arm has to take place. This is called a depth scan. The depth scan is produced by a scan of the reference mirror mounted on to a piezo actuator. The maximum length of the depth scan is $500\text{ }\mu\text{m}$.

During the depth scan, an intensity variation occurs in each pixel of the CCD camera. This variation is called a correlogram. The maximum contrast of the correlogram defines the location of equal path lengths. The data evaluation has to detect the location $z_0(x,y)$ of equal path lengths for each pixel (x,y) . There are many methods to evaluate correlograms described in the literature [4]-[11]. All of the pixels make up a point cloud of the object surface.

This standard interferometer head is integrated into the measurement system as shown in Fig.2. The measurement head is mounted on a translation stage. The light of the object beam is split into two parts by means of a beam splitter. Each of the split beams is redirected with a mirror to one surface of the object. The translation stage gives the system the ability to focus on the surface of the object. This focusing function allows objects of different thicknesses to be measured.

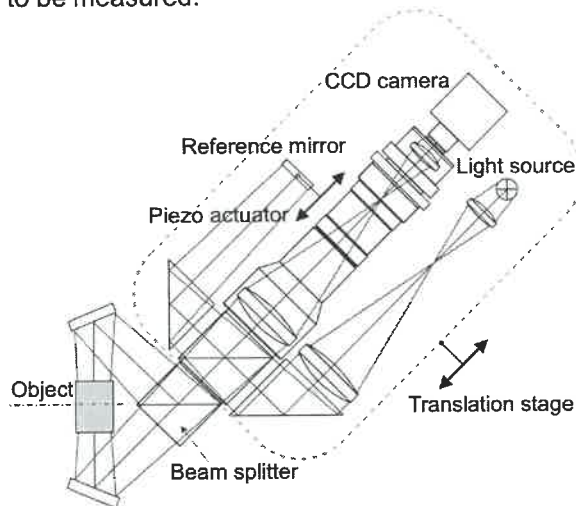


FIGURE 2. Optical setup of an interferometrical system used for high precision measurements of flatness, thickness and parallelism.



FIGURE 3. Technical realization: FTP40.

Both sides of the object are projected onto the camera simultaneously which allows the full lateral resolution of the camera and imaging lens to be achieved for both sides of the object. To separate the two sides of the short coherent feature, the measurement head is used in such a way that the signals for each side are separated by a defined distance as can be seen in Fig. 4. This is done by slightly offsetting the object from center in its set up position.

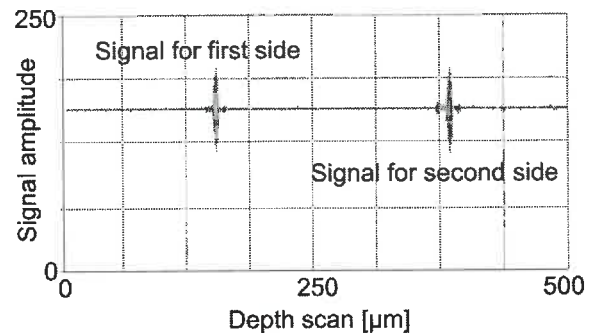


FIGURE 4. Intensity of signal over the complete 500 µm piezo scan-range

EXAMPLES OF MEASUREMENTS

A typical mechanical part to be measured with FTP40 system is shown in the Fig.5. The plate has a diameter of 30 mm and the surfaces on both sides of the part have a spherical shape with a height of 5 µm.



FIGURE 5. Typical mechanical part to be measured with the FTP 40.

Results of a 3D measurement from both surfaces of the plate are shown in the Fig. 6 to Fig. 8.

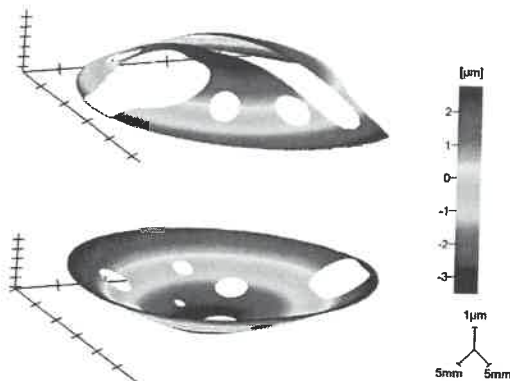


FIGURE 6. 3D measurement of an example plate.

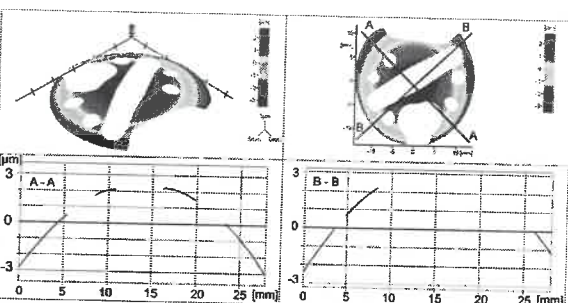


FIGURE 7. 3D shape and profile of an identified cross section on the top surface of the plate.

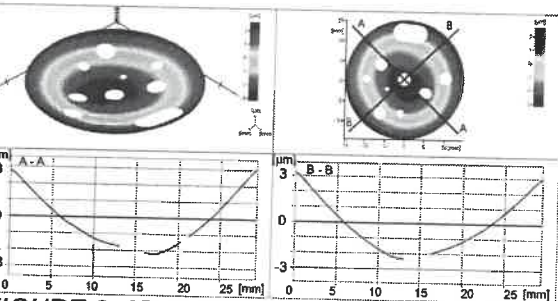


FIGURE 8. 3D shape and profile of an identified cross section on the bottom surface of the plate.

The plate was used to perform reproducibility test measurements of flatness, thickness and parallelism. In this case "flatness" of the spherical surface was calculated as the difference between the actual and nominal spherical form. Thickness of the plate was defined as a distance between two parallel planes which were defined from the top and bottom of the plate. Parallelism was calculated from two best fit planes.

Results of 25 repeatability measurements are shown in Fig.9 to Fig.11.

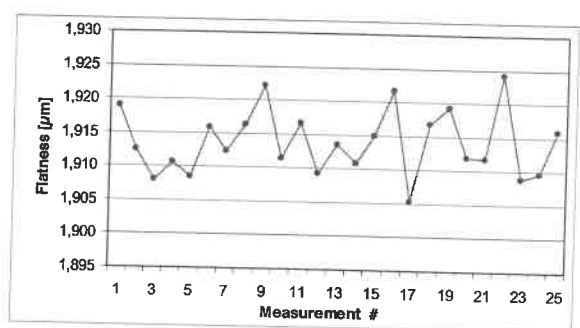


FIGURE 9. Results of 25 flatness measurements.

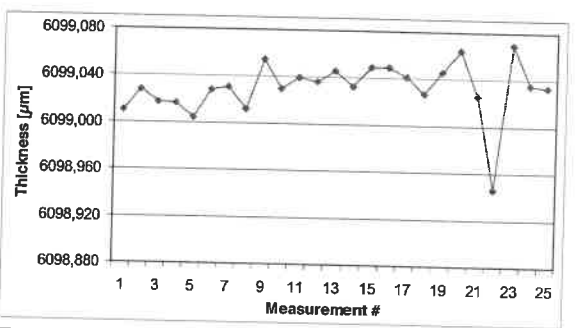


FIGURE 10. Results of 25 thickness measurements.

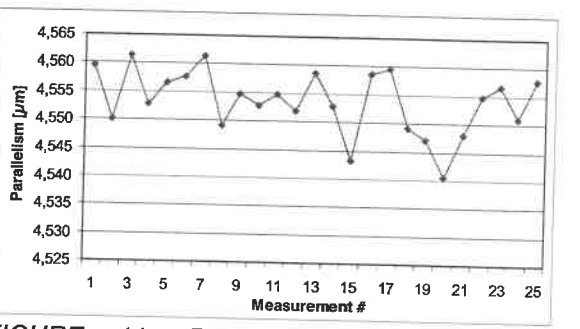


FIGURE 11. Results of 25 parallelism measurements.

The standard deviation of each series of 25 measurements was calculated based on following a Cg procedure [3] in which the measurement systems capabilities were evaluated based on the following tolerances:

- Spherical form ("flatness") $T \geq 0,215 \mu m$
- Thickness $T \geq 1,050 \mu m$
- Parallelism $T \geq 0,210 \mu m$

Another typical task for the FTP40 system is the measurement of distances between multiple components on one device. Results of an example distance measurement between three components are shown in Fig.12.

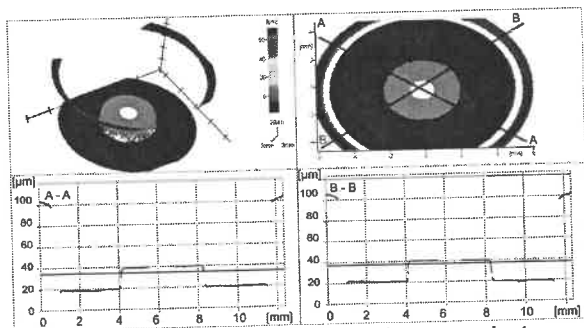


FIGURE 12. Measurement of distances between plains of three components.

CONCLUSION

An interferometrical system which allows flatness, thickness and parallelism measurements of precision mechanical parts with tolerances less than $1\mu\text{m}$ has been developed, built, tested and already used under shop floor conditions.

The optical system is based on a concept of conventional Michelson white light interferometry with a depth resolution of 1 nm . One basic advantage of white light interferometer is that the depth resolution is independent of the aperture of the objectives. The high resolution in depth is necessary in order to perform measurements of form of mechanical parts which have tolerances of less than $1\mu\text{m}$.

The interferometer will be used for fast measurements of flatness, thickness and parallelism either simultaneously or individually. The measuring head is situated on displacement table making it possible to measure parts from 0 to 100 mm in height or to measure distances between two surfaces on any one part. To maximize the precision of the measurement, a gage measurement would be made as a reference before each part measurement. As a result, the actual part measurement data can be compensated from the known error of the system.

Typical parts are mechanical components such as distance plates, bodies, gage blocks and optical plates. The stable mechanical construction of the system allows for possible automation by using custom fixtures or possibly implementing a robot for auto part loading.

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REAL-TIME PERIODIC ERROR COMPENSATION WITH LOW/ZERO VELOCITY PARAMETER UPDATES

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INTRODUCTION

It has been shown recently that periodic errors caused by frequency leakage in displacement measuring heterodyne interferometers can be compensated in real time [1]. The approach is to acquire error parameters during target motion in 1 millisecond intervals and apply the current error parameters through digital compensation in the next millisecond, while again measuring the error parameters. In this way, the compensation is applied in a leap-frog manner with 1 millisecond latency. While compensation can be applied at any travel velocity, updating the error parameters currently requires that a minimum travel speed be met or exceeded. There are applications, however, where this minimum speed is seldom or never reached in operation. Therefore, the parameters cannot be updated even though they are continuously changing. An example of this situation is the x-axis interferometer on a stage moving mainly in the y -direction (such as the position feedback for a lithographic stepper machine used in semiconductor fabrication). Mirror imperfections cause beam shear in the x-axis interferometer, which alters its periodic error magnitude even though there is effectively no x -travel.

In this work, it is shown that by monitoring the interference signal strength, it is possible to update first order periodic error magnitude even at low/zero velocities because there is an inverse-proportional relationship between the v and Δ . A setup is presented where beam shear is intentionally modified to vary the intended interference signal magnitude, while maintaining the first order periodic error signal magnitude. The resulting variation in first order periodic error magnitude is identified from the position signal.

BACKGROUND

A schematic of a single pass heterodyne displacement measuring interferometer is

provided in Fig. 1. The two light frequencies, f_1 and f_2 , are collinear, but orthogonally polarized so that they can be separated at the polarizing beam splitter (PBS). Ideally, f_1 travels only to the moving target and f_2 only to the fixed target. However, due to physical imperfections, frequency leakage occurs. Specifically, f_1 and f_2 travel to both the moving and fixed targets. In this case, rather than obtaining only the intended interference term (f_1 to moving target and f_2 to fixed), multiple interference terms appear in the frequency spectrum. These multiple "interferometers" yield the cyclical perturbations in the measurement signal referred to as periodic error. The frequency leakage is depicted in Fig. 1 using an asterisk to identify the unintended frequency in each path.

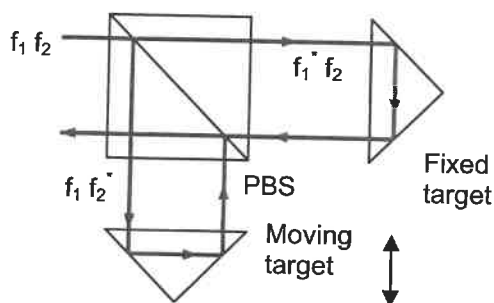


FIGURE 1. Schematic of single pass heterodyne interferometer with frequency leakage.

The individual terms that appear in the frequency spectrum for the fully leaking interferometer are: 1) $f_1 - f_2$, intended signal; 2) $f_1 - f_2^*$ and $f_1^* - f_2$, responsible for first order periodic error; 3) $f_1 - f_2^*$, responsible for second order periodic error; and 4) $f_1 - f_1^*$ and $f_2 - f_2^*$, equivalent to homodyne interference signals (only problematic at high velocities and corresponding large Doppler shifts [2]).

Beam shear, or a change in overlap between the beams returning from the fixed and moving

targets, is introduced by a lateral (left/right) offset of the moving target. This is demonstrated in Fig. 2. In this case, the magnitude of the first order periodic error signals are unaffected because they exist in a single path; both f_1^* and f_2^* appear in the fixed path and both f_1 and f_2 appear in the moving path. However, the magnitudes of the remaining signals do vary with the change in beam overlap. For example, the magnitude of the intended interference signal between f_1 and f_2 varies with beam shear because the overlap between the two (nominally) Gaussian profile beams changes.

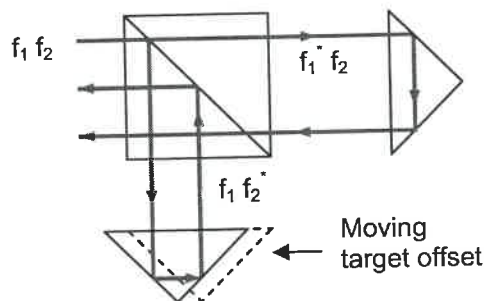


FIGURE 2. Beam shear caused by lateral motion of the moving target.

Periodic error magnitudes can be calculated using the magnitudes of the individual interference terms [3-4]. First order error depends on the ratio of the $f_1^*-f_2/f_1-f_2^*$ signals, which appear at the beat frequency (i.e., the difference between the f_1 and f_2 frequencies), and the f_1-f_2 intended interference signal, which appears at the beat frequency shifted by the Doppler frequency (that varies with the target velocity). Because the magnitudes of the $f_1^*-f_2/f_1-f_2^*$ signals are constant, but the intended interference signal magnitude is not, their ratio varies and first order periodic error magnitude changes with beam shear.

Second order error depends on the ratio of the $f_1^*-f_2^*$ signal, which is also Doppler shifted from the beat frequency but in the opposite direction from the intended interference signal, and the f_1-f_2 intended interference signal. Because both signals are attenuated with beam shear, the ratio remains approximately constant and the magnitude of second order periodic error does not change with beam shear.

As noted, however, these error magnitude calculations require that the individual signals can be resolved in the frequency spectrum,

which demands relatively high target velocities to achieve the necessary separation via the Doppler shift. In this work, it is shown that low velocity compensation for first order periodic error is possible due to the relationships just described.

EXPERIMENTAL SETUP

In order to enable variable beam shear in a single pass heterodyne interferometer, a dedicated setup was developed; see Fig. 3. In this setup, a two-frequency laser head directs the 1.5 mm diameter, collinear light beams first through a non-polarizing beam splitter (NPBS), where the reference signal for phase measurement is obtained. The beams then pass through an adjustable half wave plate (HWP) which enables the linear polarization vectors to be rotated and changes the amount of frequency leakage in the PBS. Next, the two frequencies are nominally separated at the PBS and travel to the fixed and moving targets; the moving retroreflector (RR) is mounted on an air bearing stage. The beams are recombined in the PBS, turned 90 deg by a prism and pass through an adjustable linear polarizer (LP), which enables the interference signal magnitudes to be modified. This provides the measurement signal for the phase measuring electronics (312.5 kHz sampling frequency, 0.3 nm resolution).

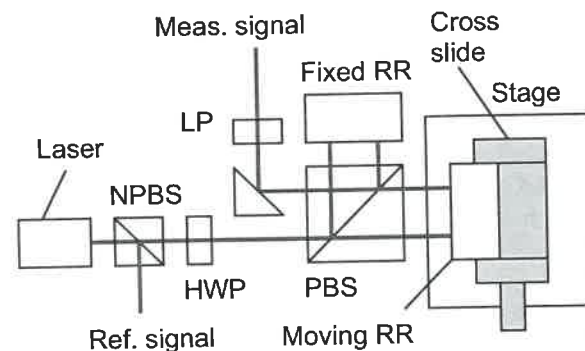


FIGURE 3. Schematic of experimental setup to enable adjustable beam shear.

In the setup, the axis of the air bearing stage was first carefully aligned with the beam axis the alignment was verified by monitoring the magnitude of the interference terms at high velocity along the full stage travel using an analog spectrum analyzer. Beam shear was then controlled by adjusting the lateral position of the moving RR using a micrometer cross slide. Tests are then conducted by setting the

cross slide position for a particular beam offset, collecting the position data using the phase measuring electronics, and then analyzing the data to determine the first and second order periodic error magnitudes. The periodic error magnitudes were identified by: 1) separating the position data into multiple sections; 2) subtracting a least squares polynomial fit to remove the macroscale motion and isolate periodic error; 3) calculating the discrete Fourier transform of the residual time-domain data; 4) identifying the first and second order periodic error magnitudes based on the Doppler shift (commanded velocity); and 5) averaging the results from the individual sections.

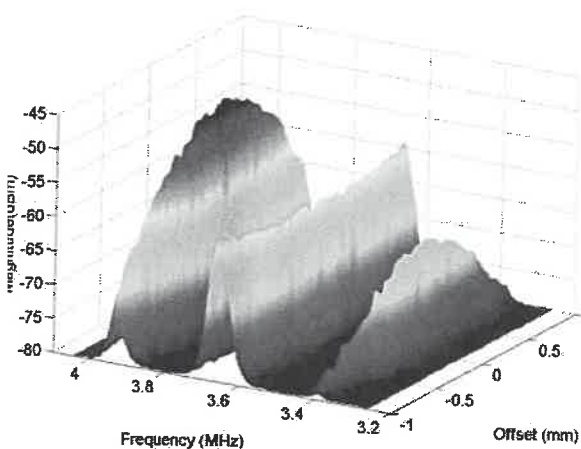


FIGURE 4. Variation in signal magnitudes with changes in the moving RR offset (beam shear).

EXPERIMENTAL RESULTS

Figure 4 demonstrates the variation in signal magnitudes with changes in offset. The HWP and LP angles were 10 deg from their nominal orientations to introduce significant periodic error content. The data was collected using a spectrum analyzer and the velocity was 5000 mm/min to produce sufficient frequency separation. It is seen that the intended interference signal (left) magnitude varies significantly with offset. It is maximum when the beams from the fixed and moving RRs are exactly overlapped (zero offset) and reduces with the moving RR lateral offset in either direction (±1 mm in the plot). The situation is similar with the second order error signal (right) magnitude. However, the magnitude of the first order error signal (middle), which appears at the 3.6 MHz offset frequency, does not change with offset as discussed previously.

Figure 5 shows the variation in first and second order periodic error magnitudes with offset. These magnitudes were determined from position data (obtained using the phase measuring electronics) with 3 deg HWP and LP misalignments and a velocity of 100 mm/min. Figure 6 shows the results for 5 deg HWP and LP misalignments. In both cases, the expected variation in first order periodic error magnitude with offset is observed. Additionally, the second order error magnitude is approximately constant, as discussed.

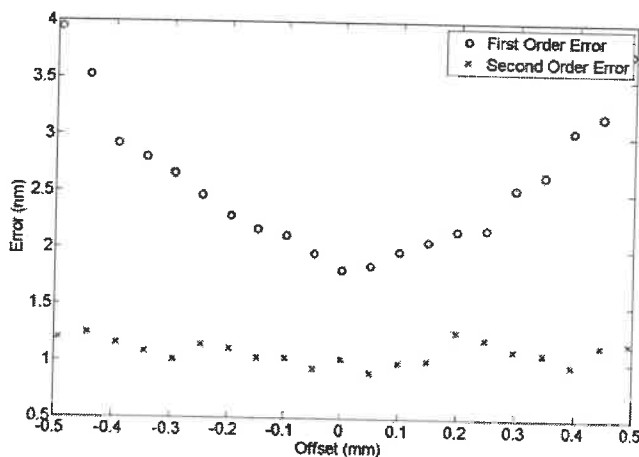


FIGURE 5. Variation in periodic error magnitudes versus moving RR offset with 3 deg HWP and LP misalignments and a velocity of 100 mm/min.

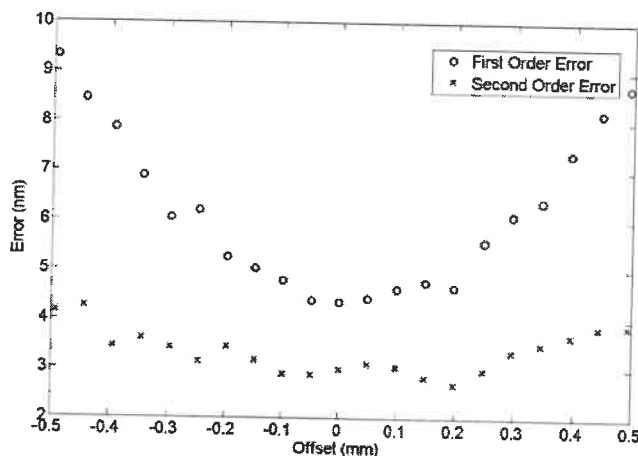


FIGURE 6. Variation in periodic error magnitudes versus moving RR offset with 5 deg HWP and LP misalignments and a velocity of 100 mm/min.

In order to demonstrate the inverse-proportional relationship between the intended interference signal and first order periodic error magnitudes,

the following procedure was completed. First, the magnitudes of the interference signals were measured at offsets from $\{+1 \text{ to } -1\}$ mm using a spectrum analyzer. A velocity of 5000 mm/min was used to provide adequate frequency separation and a misalignment angle of 10 deg was selected for both the HWP and LP to yield large periodic error magnitudes. Second, the first order periodic error magnitude was calculated using the spectrum analyzer data for each offset. The procedure described in Ref. [4] was applied. Third, a scaling factor, SF_i , was defined at each offset i which depended on the ratio of the intended interference signal magnitude, IIS_i (in dBm), at that offset to the intended interference signal magnitude at zero offset, IIS_0 . See Eq. 1, where the power relationships are required to convert from dBm to linear magnitude units. Fourth, the offset-dependent scaling factor was multiplied by the first order periodic error values. The results are displayed in Fig. 7.

$$SF_i = \frac{10^{\frac{IIS_i}{20}}}{10^{\frac{IIS_0}{20}}} \quad (1)$$

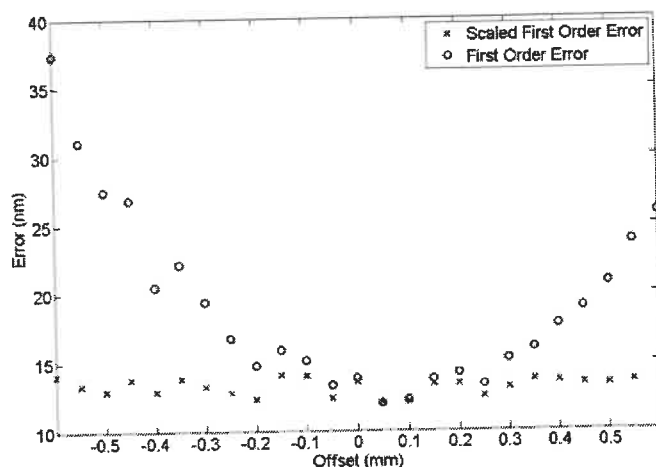


FIGURE 7. First order periodic error versus moving RR offset for 10 deg HWP and LP angles and 5000 mm/min velocity. The scaled error is also shown.

In Fig. 7 it is seen that the first order periodic error (circles) again follows the parabolic trend observed in the position data analyses from Figs. 5 and 6. The errors are large due to the significant misalignments in the HWP and LP angles. Once the scale factor from Eq. 1 is

applied at each offset, however, the scaled first order periodic error (crosses) is approximately constant with offset. This clearly identifies the inverse-proportional relationship between the intended interference signal and first order error magnitudes. Further, it shows that, provided the first order error signal magnitude can be identified (by a high velocity test), monitoring the interference signal magnitude enables the first order periodic error to be compensated, even for low/zero velocity motions.

CONCLUSIONS

In this paper, the variation in first order periodic error magnitude with beam shear was demonstrated. It was shown that by varying the overlap between the beams from the moving and fixed targets, the first order error varied parabolically and the second order error was constant. The outcome is that by first measuring the first order periodic error signal magnitude and then monitoring changes in the interference signal strength, it is possible to update first order periodic error magnitude even at low/zero velocities.

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COMPACT, NANOMETER-RESOLUTION, FIBER-OPTIC ENCODER

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INTRODUCTION

High-precision industries require positioning in multiple degrees of freedom with increasingly faster data rates and higher resolution. Recent trends make compact solutions for position measurement desirable, along with vacuum compatibility, immunity to EM interference, safety from electric discharge, minimal thermal noise and long cable lengths between readhead and controller. We have developed a new type of nanometer-resolution encoder that addresses the above requirements. The compact, high-resolution readhead contains only passive optical components: it comprises simply a grating mask covering a fiber bundle mounted in a ferrule. The monolithic design supports vacuum compatibility. Elimination of electronics from the readhead removes thermal noise and gives EM immunity. Optical fibers allow 100m+ cable lengths and high signal rates. Our unique technology enables a readhead diameter of less than 3mm, making possible one of the world's smallest linear encoders. We describe the principle of operation, design, specifications, test system and preliminary test results.

PRINCIPLE OF OPERATION

Figure 1 shows the interferometric principle of operation [1-2]. A fiber bundle consists of 6 multimode fibers (MMF) wrapped around a single mode fiber (SMF) that provides a point source of laser light of wavelength λ . The light is incident on a reflective scale that resides at a gap g from the readhead. The scale phase grating of period p eliminates 0th order diffraction, so that only diffractive orders $m = \pm 1$ interfere at the mask grating to create fringes of period p .¹ The use of a phase grating allows a large gap tolerance.

By contrast, an amplitude grating allows the 0th order and relegates fringes of period $2p$ to gaps $g = 2n\lambda/p^2$, $n=1,2,\dots$ spacing [3].

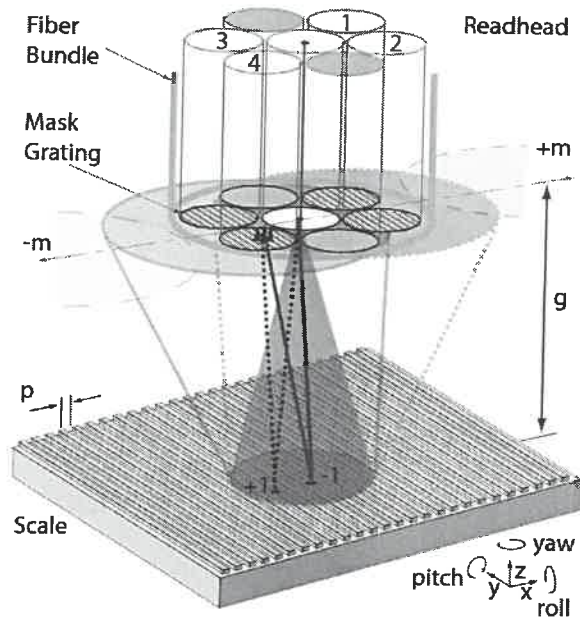


FIGURE 1. Operating principle for fiber-based encoder. Rays +1 (dotted line) and -1 (solid line) from diffraction orders $m=+1$ and -1 interfere at the mask grating of MMF Receiver 4 to create fringes of period p .

As the scale moves along the x-axis direction, each mask grating of period p transmits a sinusoidal optical signal. The relative phases of the mask gratings allow generation of analog quadrature signals. Table 1 shows some of the possible phase configurations [4] that we tested. The 4-phase configurations leave two fibers free for other uses (power monitoring, alignment, or reference mark detection).

Interpolated X location can be computed using

$$x = \frac{p}{4\pi} \text{atan2}(Q_A, Q_B) \quad (1)$$

where, for Configuration 1 in Table 1, quadrature signals are given by laterally differential signal combinations of receiver channels:

TABLE 1. Phases (in degrees) detected for various configurations of fiber receiver channels.

Receiver channels*	Configuration		
	1	2	3
CH 1	0	0	0
CH 2	180	90	120
CH 3	90	270	120
CH 4	270	180	0
CH 5	-	-	240
CH 6	-	-	240

*CHs 1-4 are labeled in Figure 1.

$$\begin{aligned} Q_A &= CH1 - CH2 \\ Q_B &= CH3 - CH4. \end{aligned} \quad (2)$$

A vector combination of the above signals (which we call Quadrature Lateral Differential Vector Composition or QLDVC) gives alternative quadrature signals [5]:

$$\begin{aligned} Q_A &= CH1 - CH2 - CH3 + CH4 \\ Q_B &= CH1 - CH2 + CH3 - CH4. \end{aligned} \quad (3)$$

Testing shows that Configuration 1 with QLDVC signals achieves best misalignment tolerances and robustness to scale contamination.

SIMULATION

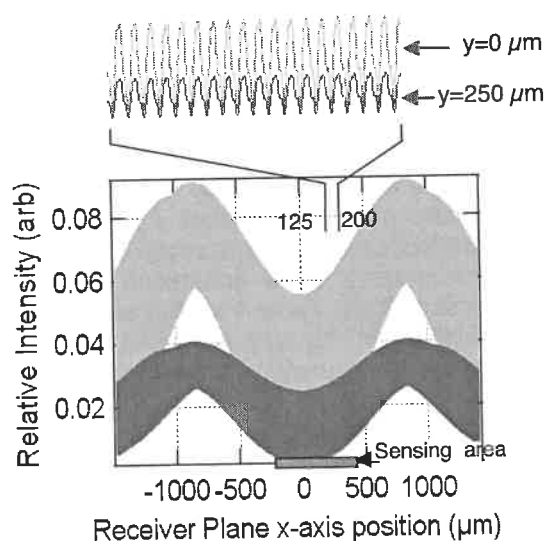


FIGURE 2. GLAD simulation [6] of readhead intensity profile in receiver XY plane. The fringe period is 4 μm, giving a signal period of 2 μm.

We performed physical optics simulation of the field intensity at the mask-grating plane of the readhead. Figure 2 displays the resulting intensity profile when using the parameters in Table 2. The simulation shows high contrast and insignificant fringe position aberration (<0.2 μm total) across the readhead sensing area.

TABLE 2. Readhead design parameters

Parameter	Value
Gap g (mm)	5
NA, SMF	0.11
Receiver Diameter (mm)	0.25
Receiver core diameter (mm)	0.20
Fiber to fiber spacing (mm)	0.25
Optical wavelength (μm)	0.65
Scale grating period p (μm)	4
Nominal scale step height (μm)	0.164
Nominal scale duty cycle DC (%)	50
Mask grating period (μm)	4

COMPONENTS

Figure 3 shows a scale fabricated at Mitutoyo and typical readhead components made by standard optical fiber and telecom industry processes. Passive alignment of mask is performed under microscope with XYZ-yaw adjustments (approximately 10 μm and 0.2° tolerances).

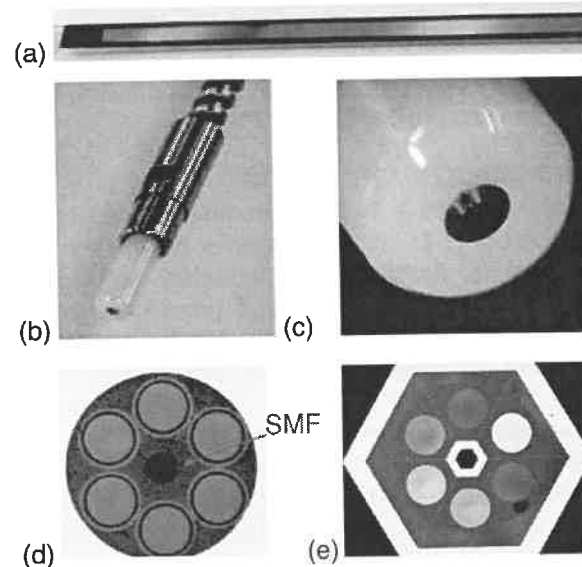


FIGURE 3. Components: (a) scale; (b) readhead fiber bundle; (c) end of ferrule; (d) close-up of fibers glued into ~750 μm ferrule hole; (e) Configuration 1 mask grating mounted over ferrule with backlit fibers.

TESTING

We test prototype systems with a hexapod-based system (Figure 4) under control of LabVIEW software that automates multiple degree-of-freedom testing in step-and-repeat mode. Acquired data includes reference position, receiver signals, QLDVC quadrature signals and temperature. We convert quadrature signals to position and do further analysis offline.

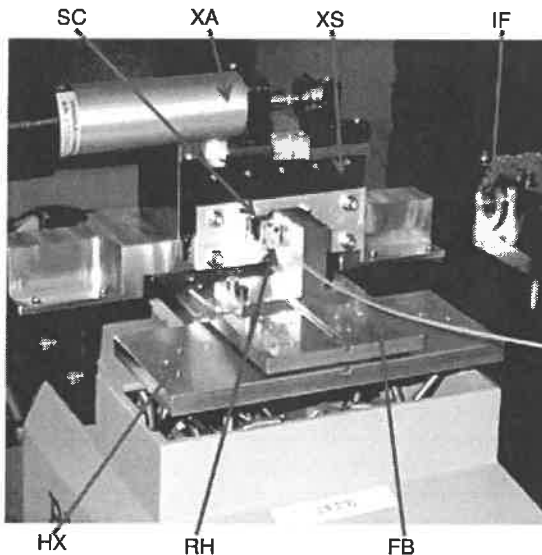


FIGURE 4. Test platform. Readhead (RH) with attached fiber bundle (FB) is fixed on hexapod stage (HX). Encoder scale (SC) is mounted to X-axis stage (XS) that bridges the hexapod and is driven by actuator (XA). Interferometer (IF) obtains a reference position. Temperature is controlled to 0.05° using a custom enclosure.

Typical test data at best readhead alignment are shown in Figures 5 and 6. The test system is currently limiting measured performance. Improvements are under way.

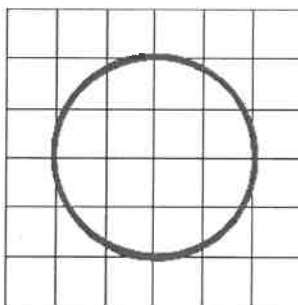


FIGURE 5. Lissajous figure of QLDVC signals (Q_A vs Q_B). Each division is $0.5V$.

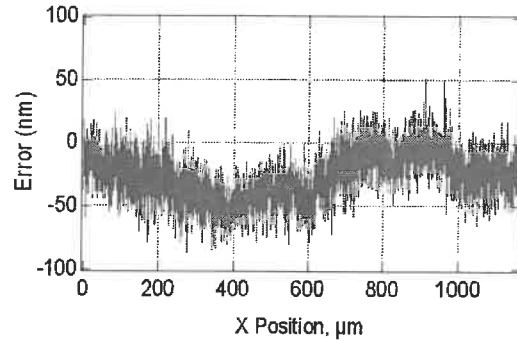


FIGURE 6. Position error. Standard deviation (1σ) is 19 nm , or 13 nm with long range drift ($100\text{ }\mu\text{m}$ smoothing window) removed.

Figure 7 shows typical signal levels for various readhead misalignments against a 90% signal strength requirement. Table 3 summarizes test results.

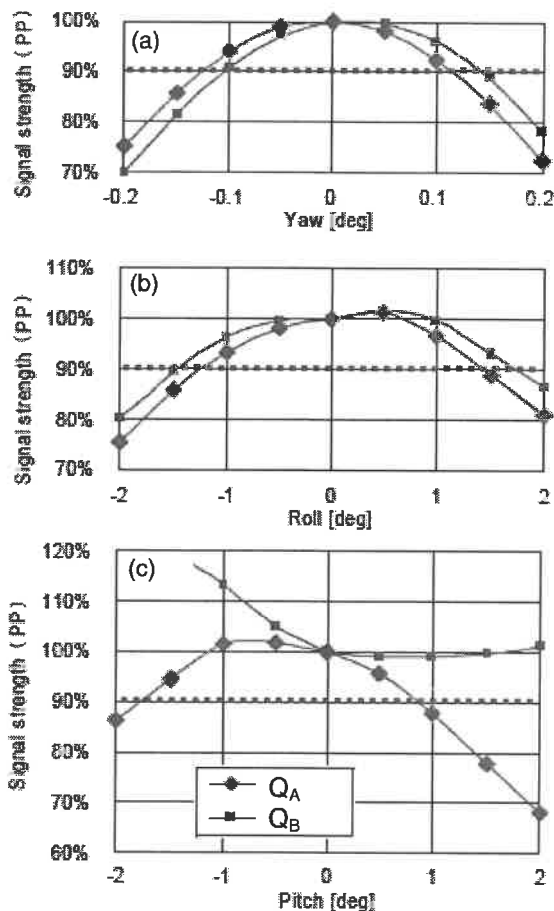


FIGURE 7. Signal strength for output analog quadrature signals with (a) yaw, (b) roll and (c) pitch misalignment. The 90% signal level requirement is shown as a dashed line.

TABLE 3. Typical test results.

Parameter	Result
Misalignment Tolerances*	
Gap (mm)	2 ± 0.2
Pitch ($^{\circ}$)	0 ± 1.2
Roll ($^{\circ}$)	0 ± 1.6
Yaw ($^{\circ}$)	0 ± 0.11
Position error (nm)	$< 20 (1\sigma)^{**}$

*Within which 90% signal level is maintained.

**For 1.2mm length of scale.

SYSTEM DESIGN AND SPECIFICATION

Figure 8 shows the system design and Table 4 gives target specifications. Readhead configurations include both straight sensing (shown) and side sensing (with use of a turning prism). An OEM readhead with diameter $< 3\text{mm}$ is possible.

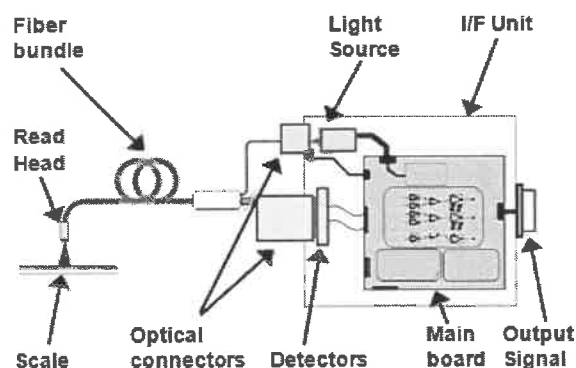


FIGURE 8. Encoder system comprising scale, readhead, armored-cable fiber bundle with optical connectors, and interface unit (I/F). The I/F comprises laser diode (LD), receiver PIN PD detectors, and analog circuitry for LD control (driver and power stabilization), alignment, and QLDVC signal processing with offset, amplitude and phase adjustment to produce analog quadrature output signal. Interpolation is performed separately.

CONCLUSION

The optical fiber based encoder offers a compact, high resolution readhead with large misalignment tolerances and unique performance characteristics such as EM immunity, low thermal noise and long cable lengths. Testing to specifications is nearing completion. Final development is under way with product launch planned for 2010. Additional product options (such as reference marks) are under investigation.

TABLE 4. Preliminary target specifications

Parameter	Requirements
Configuration	Reflective
Laser diode wavelength	655 nm
Phase grating scale	$2/2 \mu\text{m}$ line/space
	50-100 mm length
Fiber bundle lengths	2-30 m
Max Velocity	0.5 m/s
Signal pitch	$2 \mu\text{m}$
Resolution (with interpolator)	1 nm (2000X) 5 nm (400X)
Gap	$5 \pm 0.15 \text{ mm}$
Pitch and roll misalignment	$\pm 1^{\circ}$
Yaw misalignment	$\pm 0.1^{\circ}$

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TESTING QUANTUM DOTS AS A MEANS OF ASSESSING SUBSURFACE DAMAGE IN POLISHED GLASS

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SUMMARY

Fluorescent materials in the form of quantum dots (nanocrystals of cadmium-selenide) were added to the abrasive slurries used in lapping and polishing glass samples in an attempt to tag damage sites as they were created during the material removal processes (Figure 1). Quantum dots that were trapped in these defects were then subject to laser excitation and imaged on a confocal fluorescence microscope. Samples which were lapped in the presence of quantum dots showed numerous significant features, features which were not detected with optical microscopy or white light interferometry, but which were confirmed with follow up etching of the samples that revealed a fractured layer.

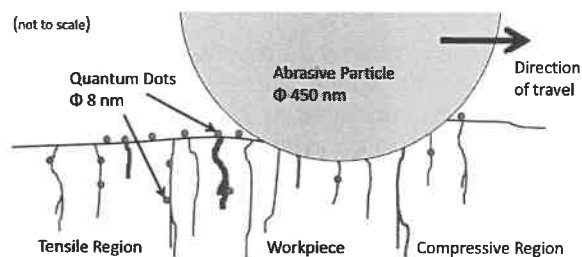


FIGURE 1. Hypothesized interaction of quantum dots with the surface near the travelling abrasive particle.

The fluorescent features observed displayed differing fluorescent responses as the depth of the focal plane increased. One type of feature showed a rapid falloff in the fluorescence as the focal depth increased, indicative of a quantum dots being trapped in near the surface. The second type of feature observed has a fluorescent response that did not diminish from the surface to a depth of 10 μm , indicating a deeper fracture containing quantum dots.

BACKGROUND

Subsurface Damage

Subsurface damage (SSD) is a layer of fractures, stressed material, and other defects that can lie beneath a polished glass surface that shows no indication of the underlying damage. SSD is thought to be a result of more aggressive material removal processes in the finishing of glass such as lapping and grinding. SSD is described as a layer of deformed material over the bulk layer is covered by a layer of fractures and defects. These defects are then covered over by a thin layer of smooth material in subsequent polishing operations [1, 2].

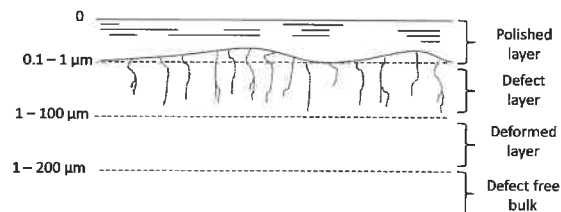


FIGURE 2. Illustration of the various SSD layers and their depths [2].

The presence of SSD in a polished component can be detrimental to the performance as it can introduce optical aberrations, reduce the fatigue strength in oscillating components, and reduce the laser induced damage threshold (LIDT), which is a measure of how much laser energy a component can tolerate without risking catastrophic failure. Defects in the subsurface layers may also propagate to the surface under stressed conditions (such as coating operations or deployment in extreme conditions).

As it lies beneath the surface, SSD cannot be detected with interferometry or contact methods, as these methods measure only the topmost polished layer. Traditionally, SSD has been evaluated by etching away the smooth polished

layer to reveal the defect layer beneath or polishing a known shape into the surface (either taper polishing or dimpling) then using geometry and microscope observations to determine the depth of the damaged layer [2, 3]. Papers by Brinksmeir [4], Lucca [5], and Shen [6] have reviewed recent techniques to assess SSD such as X-ray diffraction, laser modulated scattering and Raman spectroscopy.

Quantum Dots

Quantum dots are semiconductor nanocrystals that fluoresce with narrow emission spectra when subject to excitation by a light source. They possess several characteristics that make them more desirable than the fluorescent dyes such as higher fluorescent response, narrow emission spectra, and resistance to photobleaching (the inability to fluoresce after a series of excitations) [7].

EviDots from Evident Technologies were used as the quantum dots in these experiments. They consist of cadmium selenide cores surrounded by zinc sulfide shells (which increase the resilience of the quantum) and a layer of ligands (~2 nm thick) to improve the solubility. The particular EviDots used have a crystal diameter of 3.8 nm and an overall diameter of 7.8 nm as well as an emission peak of 553 ± 10 nm with a 40 nm full width half maximum (FWHM) breadth to the emission peak [7].

EXPERIMENTAL PROCEDURE

Sample Preparation

Glass samples were lapped for twenty minutes by hand on an iron platen with an abrasive slurry containing 20 μm alumina particles to which a mixture of quantum dots in toluene were added. The samples were immersed in a quantum dot solution for one hour, then polished on a Dacron cloth pad for 30 minutes with an abrasive slurry containing 0.45 μm ceria particles as well as quantum dots in solution. Prior to imaging, the samples were wiped clean with isopropyl alcohol soaked tissues, a process shown to remove quantum dots adhered to the glass surface. Details on these finishing and cleaning processes can be found in the author's earlier report on this work [8].

The glass samples also underwent a 5 minute pitch polishing process on an Acculap™ Standard pitch tool. A 0.45 μm ceria abrasive slurry was used in the pitch polishing, but the

slurry did not contain any quantum dots. This step was performed to see if polishing dynamics would remove any quantum dots that were embedded in the surface, but not trapped in the subsurface. AFM measurements had shown spots on the sample surface that occurred with a similar frequency to features present in the confocal fluorescence images [8]. This pitch polishing step was estimated to have removed 250 nm from the surface (based on mass loss) and dramatically reduced the incidence of low intensity fluorescent sites on the surface.

The samples were examined with a Mitutoyo Finescope at magnifications of 50x, 100x, and 200x with very low incidence of scratches or pits. The sample roughness was measured with a Zygo NewView White Light Interferometer to be 1 nm R_a over a 80 μm $\times$ 110 μm scan.

Confocal Fluorescence Imaging

The glass sample is mounted on a cover slip and placed in a custom inverted confocal microscope as described in detail by the author in an earlier publication [9]. A confocal microscope differs from a conventional microscope in that the scan area is illuminated and imaged point by point instead of all at once. This allows the confocal microscope to reject fluorescence from out of focus planes and adjacent points.

The sample is moved during the 40 μm $\times$ 40 μm scan while the optics remain stationary. The stage is then translated in the Z-axis and a new 40 μm $\times$ 40 μm scan is taken at a new focal plane. Excitation is provided by a 470 nm diode laser, while the fluorescence is measured by a single photon avalanche diode, which is filtered by a 538 nm long pass filter (to reject any reflected light from the laser). The fluorescence counts are collected by the Nanoscope software controlling the stage. The data is then exported for analysis in MATLAB, where the fluorescent signals are normalized with a relative fluorescence of one being equal to the maximum fluorescence observed on a sample without any quantum dots. Thus a relative fluorescence much greater than one corresponds to a significant number of quantum dots within the excitation volume of the confocal microscope at that position.

Etching to Reveal SSD

The glass sample was suspended in a dilute (2%) solution of hydrofluoric acid for varying

urations. The samples were examined with an optical microscope for any signs of fractures and defects. Sample masses before and after each etching step were used to estimate the depth of material removed.

RESULTS

Confocal Fluorescence Results

Significant fluorescence was detected on the samples which had been lapped and polished with quantum dots, then pitch polished for 5 minutes without quantum dots. While the vast majority of the scan area registered fluorescence values within the background (relative fluorescence ≤ 1), there were features with relative fluorescence more than an order of magnitude greater than the background (Figure 3 and 4).

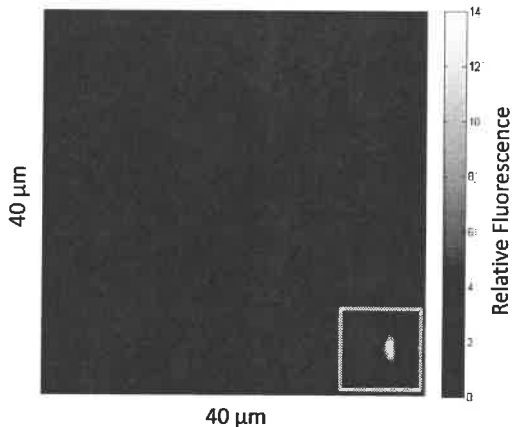


FIGURE 3. Relative fluorescence in a confocal scan of a glass sample lapped and polished in the presence of quantum dots, then pitch polished for 5 minutes (taken at the surface).

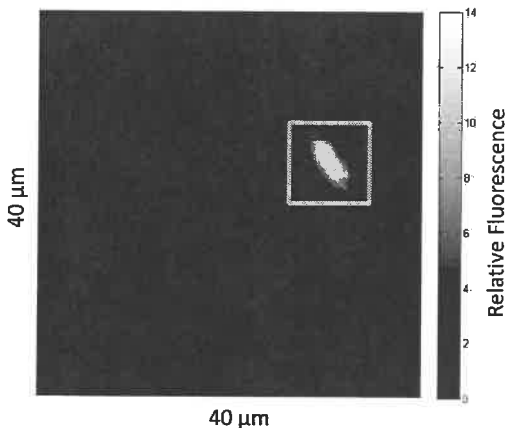


FIGURE 4. Confocal scan of a different surface location on the same sample shown in Figure 3.

While the fluorescent images at the surface look similar between the locations shown in Figure 3 and Figure 4, the fluorescent response as the focal plane moved deeper into the sample were quite different. The intense fluorescent feature in the bottom right corner of Figure 3, quickly diminished in fluorescence as the focal plane went deeper into the sample (Figure 5). This drop off in fluorescence is similar to that seen in slides with quantum dots dried on the surface, where the maximum fluorescence drops by 40%-60% when the focal plane is 10 μm below the surface.

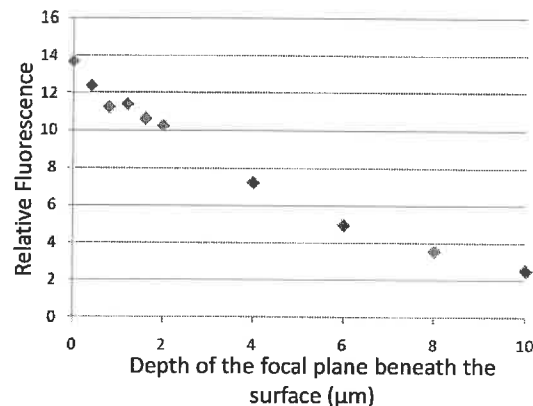


FIGURE 5. Maximum relative fluorescence in the confocal scan as focal plane depth increases at the boxed location shown in Figure 3.

The large fluorescent feature in Figure 4 had a relative fluorescence at the sample surface comparable to the feature discussed in Figure 3. The fluorescence in this feature however did not diminish as the depth of the focal plane beneath the surface reached 10 μm (Figure 6).

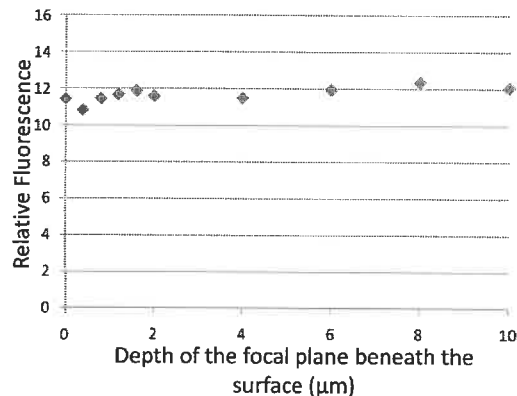


FIGURE 6. Maximum relative fluorescence in the confocal scan as focal plane depth increases at the boxed location shown in Figure 4.

Etching Results

Examination of the etched samples with the Mitutoyo Finescope showed a definite fractured layer beneath the polished surface. Many of the defects observed were comparable in size and occurrence to the fluorescent features observed in the confocal scan. These fractures remained distinct to a depth conservatively estimate to be 3 μm to 7 μm .

CONCLUSIONS AND ONGOING WORK

Adding quantum dots to the abrasive slurries used in lapping and polishing the glass samples did tag features in the subsurface that were not detectable by optical microscopy or white light interferometry. The fluorescent responses observed show indications of different distributions of quantum dots in defects. The first distribution with its rapid falloff of fluorescence as focal depth increased matches previous observations of quantum dots trapped on the surface or in the immediate vicinity of the surface. The second distribution with the persistent fluorescence as focal depth increases, suggests that quantum dots are still in focus, which would only be possible if they were trapped in a defect that extended that deep.

In ongoing work, the fluorescence profiles of additional samples are being review to gauge the prevalence of the two types of features. Additional samples are undergoing etching and fracturing to obtain a better estimate of the depth of SSD to see how this correlates with the observed fluorescent responses.

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CREATING EXTREME SUBSURFACE INTEGRITY BY LASER-INDUCED MATERIAL SELF-ORGANIZATION

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INTRODUCTION

Single-crystal silicon and germanium are not only dominant semiconductor materials but also important infrared optical materials. The manufacturing of defect-free substrates of silicon and germanium is indispensable for production of precision dark-field optics, optoelectronic elements, and micro electromechanical systems. Usually, silicon and germanium substrates are manufactured by mechanical machining processes such as slicing, cutting, grinding and lapping followed by polishing. However, these processes will inevitably cause subsurface damage to the crystalline structures due to the mechanical contacts between the tools and the workpieces. The subsurface damage includes a microstructural change, namely amorphization, in the near-surface layer, and a dislocation layer beneath the amorphous layer [1-3].

Removing the subsurface damage has become a subject of concentrated research interests from multidisciplinary research communities and industries. Currently, chemical etching and/or chemo-mechanical polishing (CMP) are carried out after the machining processes to remove the damaged layers. However, due to the poor controllability in processing depth, deterioration of form accuracy becomes a big problem. Other issues, such as increase in production cost, and environmental pollution due to the chemical waste fluids, are also serious problems. Instead of removing the damaged surface layer, an alternative method currently being considered is the use of laser irradiation to repair the subsurface damage [4]. The purpose of the present work is to explore the possibility of generating perfect single crystalline structures on grinding-damaged semiconductor substrates by nanosecond-pulsed laser irradiation.

PROCESSING MECHANISMS

The processing mechanism is schematized in Fig. 1. As amorphous silicon has a remarkably

higher absorption coefficient of laser light than crystalline silicon, there will be sufficient absorption of laser in the near-surface layer to form a thin liquid silicon film. The liquid layer is metallic and has a much higher absorption rate, and thus becomes thicker and thicker. The top-down melted liquid phase finally extends below the deeper dislocated region. During the period of melting, an initially rough surface becomes a smooth one due to the surface tension effect of the liquid layer. This is similar to the manner in which a free droplet of liquid naturally assumes a spherical shape to achieve a minimum surface area to volume ratio. For a plane wafer, the surface area reaches a minimum when the liquid thin film becomes completely flat at the surface. After the laser pulse then, environmental cooling will result in a bottom-up epitaxial regrowth from the defect-free crystalline region which serves as a seed for crystal growth. The epitaxial regrowth mechanism in this case is different from that in laser annealing technology [5, 6], where amorphous silicon is usually sputtered on

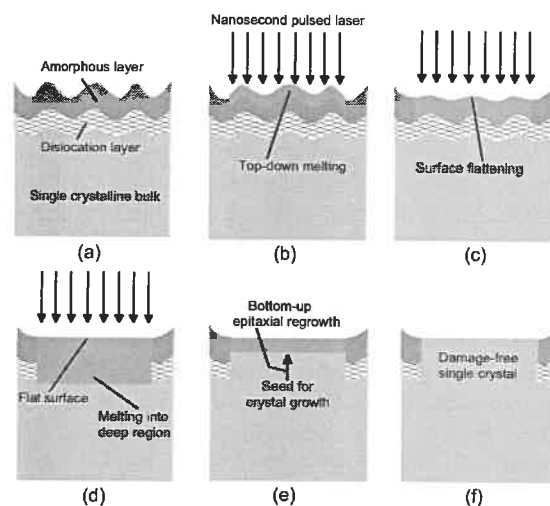


FIGURE 1. Schematic model of the processing mechanism for a single crystalline material.

substrates of different materials, such as glass or sapphire. In such cases, no lattice-matched crystal seed exists, thus poly-crystal structures will be formed. In this work, it is expected that by using the material self-organization phenomena induced by the nanosecond laser pulses, we might be able to achieve both a perfect single-crystal subsurface structure and an extremely smooth surface at the same time.

EXPERIMENTAL SETUP

In our experiments, electric device-grade p-type single-crystal silicon (100) wafers machined by diamond grinding were used as workpieces. Grinding was carried out on an ultraprecision grinder equipped with vitrified grinding wheels with fine diamond abrasives. The average size of the abrasive grains was approximately 2 μm .

The silicon wafers were then irradiated by a laser-diode excited Nd:YAG laser system, which has a wavelength of 532 nm. The laser pulse width was 20 ns, and the pulse frequency was changeable in the range of 0.1~10 kHz. The output of the laser can be adjusted by the electrical current of the laser diode. The laser output also changes with the pulse frequency. The maximum output was 1 W at a frequency of 1 kHz. The laser beam has a diameter of 600 μm at the window and has a spread angle of 0.2 mrad in air. The energy density profile within the laser beam cross-section follows the Gaussian distribution. This laser system is different from the one we used previously for processing diamond-cut samples, which had a pulse width of 3-4ns and a pulse frequency of 50 Hz [4].

The laser oscillator was mounted onto a specially developed four-axis (XYZ θ) numerical control precision stage made of ceramics (see Fig.2). The laser spot was directed to the silicon wafer which was moved horizontally by the XY tables in order to perform large-area irradiation. The XY tables of the stage were driven by linear motors on air slides, enabling a maximum speed of 0.5 m/s. The movement accuracy of the linear tables is $\sim 2 \mu\text{m}$ within the effective stroke (420 mm). The movement of the XY tables was feedback-controlled by linear scales with a resolution of 0.1 μm . The silicon wafer was vacuum-chucked on a rotary table which can rotate along the θ axis on the XY tables at a speed of 180 rpm. The rotary table has a diameter of 300 mm which enables processing large-diameter workpieces. The rotary table and the vacuum chuck were also made of ceramics and supported by precision ball bearings,

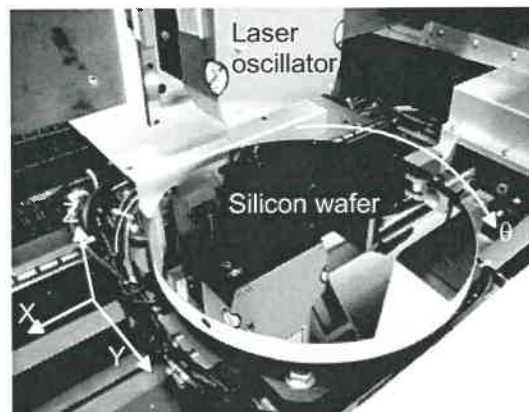


FIGURE 2. Developed experimental setup.

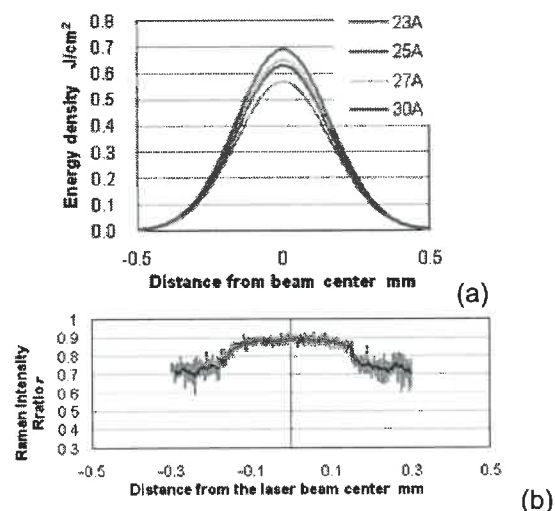


FIGURE 3. (a) Laser beam profile; (b) Raman intensity ratio change in laser irradiated region.

enabling a runout of $\sim 10 \mu\text{m}$. In the experiments, the spacing between adjacent laser irradiations was adjusted by changing the speed of the XY tables of the stage while keeping the laser frequency unchanged.

A micro-Raman spectrometer was used to characterize the microstructural changes of the samples. The wavelength of the laser used for the Raman spectroscopy was 532 nm, and the focused laser spot size was 1 μm in diameter. The laser exposure time was 1 s. In addition, cross-sectional observations of the samples were performed with a transmission electron microscope (TEM).

RESULTS AND DISCUSSION

To begin with, single-pulse laser irradiations were performed on silicon wafers. By moving the

X table of the stage at a speed of 0.3 m/s at a laser pulse frequency of 500 Hz, individual laser shots with a spacing of 600 μm were obtained. Fig. 3(a) is a plot of energy density profile of the laser beam at various electric currents in laser diode. Fig. 3(b) shows a distribution of Raman intensity ratio r [4] of a laser irradiated region. It can be seen that near the laser beam center the curve of r tends to be flat where $r > 0.85$. This region, the diameter of which is $\sim 250 \mu\text{m}$, is the laser crystallized region. By comparing (a) and (b), it is found that the critical energy density for completely recrystallize the amorphous silicon is 0.48 J/cm^2 .

To recover the subsurface damage of a large-diameter workpiece, it is necessary to scan the laser beam across the workpiece so as to overlap successive irradiation runs. Perfect crystallinity at the boundary among adjacent irradiations is essential for guaranteeing the subsurface integrity of the entire workpiece. For this purpose, the design of the scanning path of the laser beam on the wafer is an important step. The laser scanning path should meet two requirements: one is that the energy density over the entire wafer must be higher than the critical value (0.48 J/cm^2); the other is that the processing speed must be as high as possible. Fig. 4 shows plots of Raman intensity ratio of one-dimensionally overlapped irradiations at different spacing (350 and $250 \mu\text{m}$). The Raman intensity measurements were performed along the line passing through the laser irradiation centers. In the figure, the corresponding energy density profiles were also shown. In Fig. 4(a), there are remarkable depressions on the Raman intensity ratio curve between adjacent laser shots, indicating that the overlap spacing is too large to achieve complete subsurface reconstruction. In Fig. 4(b), however, the curve tends to be very flat ($r > 0.85$), demonstrating that the overlapping spacing under this condition is small enough to achieve a continuous single crystalline structure.

Next, two-dimensional overlapping irradiation was performed. Fig. 5 shows two examples of Raman intensity ratio mapping after laser irradiation at different spacing rates. Fig. 5 (a) is obtained at a spacing of $150 \mu\text{m}$ in X direction and a spacing of $300 \mu\text{m}$ in Y direction. In this case, the spacing in Y direction is too big so that residual amorphous region can be clearly seen. Fig. 5 (b) is obtained at a spacing of $200 \mu\text{m}$ in both X and Y directions. In this case, the maximum distance between any two adjacent

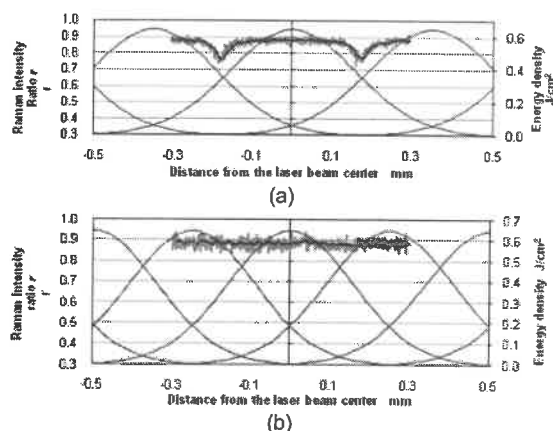


FIGURE 4. Plots of Raman intensity ratio of overlapping irradiated wafer at various spaces: (a) $350 \mu\text{m}$, (b) $250 \mu\text{m}$.

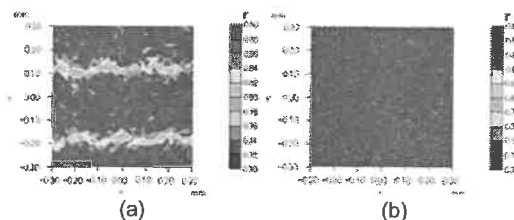


FIGURE 5. Raman intensity ratio mapping after laser irradiation at different overlapping spacing: (a) $150:300 \mu\text{m}$, (b) $200:200 \mu\text{m}$.

laser irradiations is smaller than the critical value ($250 \mu\text{m}$), so that the amorphous silicon on the entire wafer surface has been completely transformed into the crystalline silicon.

Fig. 6 (a) shows a cross-sectional TEM micrograph of the diamond-ground silicon wafer before laser irradiation. It is clear that the ground surface is far from being flat but has waviness in the size scale of a few tens of nanometers. Below the ground surface, there is a non-uniform gray layer (grinding-induced amorphous layer). Below the amorphous layer, there is a layer of dislocations. Fig. 6 (b) is a cross-sectional TEM photograph of the laser irradiated wafer at a spacing of $200 \mu\text{m}$ (XY). The amorphous layer has completely disappeared, indicating the recrystallization of silicon was performed successfully. However, there are still a few residual dislocations. Fig. 6 (c) is a TEM photograph of the laser irradiated wafer at a much smaller spacing of $100 \mu\text{m}$ (XY). The minimum laser energy density in this case is 0.63 J/cm^2 . It is clear that both the grey layer and the dislocations have completely disappeared and a perfect single crystalline structure identical to that of the bulk material has

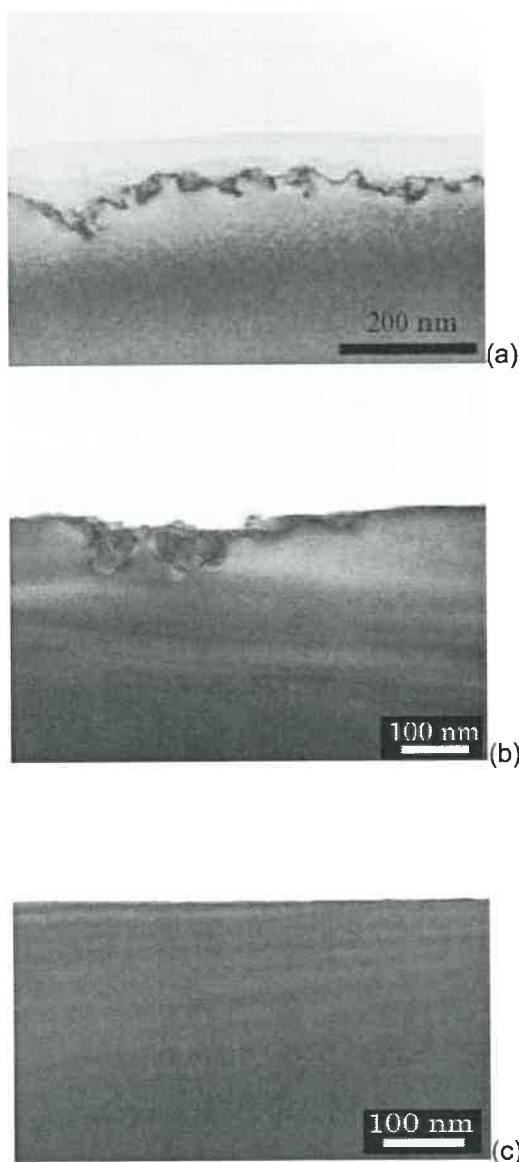


FIGURE 6. Cross-sectional TEM micrographs of silicon wafers: (a) before laser irradiation, and after laser irradiation at two-dimensional spacing of (b) $200\ \mu\text{m}$ and (c) $100\ \mu\text{m}$.

been achieved. From this result, we can say that a sufficient laser energy density is essential for achieving complete recovery of grinding damage, and that the laser energy density required for repairing dislocations is higher than that for recrystallizing the amorphous silicon. From these TEM micrographs, we can also see that after laser irradiation, the surface roughness has been remarkably improved. In other words, the grinding-induced tool marks and micro waviness on the surface have been successfully flattened. This surface flattening phenomenon might be a

result of surface tension effect of the melted silicon. That is, the surface area reaches a minimum when the liquid thin film at the surface becomes completely flat.

In addition, we performed laser irradiation tests on diamond-machined germanium substrates and confirmed similar effects. The critical laser energy density for recrystallize amorphous germanium was found to be different from that of silicon. Details will be given in the future.

The proposed laser processing technique offers a number of advantages over the conventional chemo-mechanical processes: (i) It involves no material removal thus preserves the dimensions of the workpiece; (ii) It generates no pollutants; (iii) It enables selective processing and processing of complex shapes like aspherical and diffractive optical elements.

SUMMARY

Diamond-ground silicon and germanium wafers were irradiated with a nanosecond pulsed Nd:YAG laser. The phenomena of laser-induced material self-organization were confirmed. The results indicated that if the laser energy density was sufficiently high, the grinding-induced subsurface damage was completely recovered into a perfect single-crystal structure; at the same time, the surface unevenness was remarkably smoothed. The optimum conditions for overlapping irradiation of large-diameter workpieces were experimentally obtained.

ACKNOWLEDGEMENTS

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CROSS CROWN NANOGRINDING OF A CHIPLET FOR A SLIDER WITH AN INTEGRATED MICROACTUATOR (SLIM)

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With a continuous increase in the track density, Hard Disc Drives (HDD) approach appoint where a second stage actuation will become necessary to achieve a perfect track registration. To implement such a second stage actuation in a cost competitive way, the following approach was taken: a pair of electro micromagnets were integrated into the slider of an HDD recording head, resulting in a Slider with an Integrated Micro actuator (SLIM) [2]. The micro actuator's moving part is a mounting block, carrying the read-write element located on a chiplet. Simultaneously exciting both electro micromagnets lowers the chiplet and puts the read-write element into its flying height position. On the other hand, an alternative excitation causes a minute rotation of the chiplet. As a result, a lateral displacement of the read-write element occurs. The rotation is small: for achieving a desired lateral displacement of ± 650 nm, a rotation of 0.18° (or $10.8'$) is required. While the chiplet is fabricated as an (although rather small) rectangular block, its interface to the disc requires a cross crown, else not only an appropriate chiplet location would be impossible, the chiplet edges would also dig into the disc once the chiplet is attempted to be rotated.

The chiplet's actual rotational motion on the disk dictates the required cross crown contour. For any rotation, the chiplet's trailing edge contour closest to the disc has to be tangential to the disk. Obviously, such a contour requirement is affected by chiplet and SLIM tolerances. Given the circumstances, creating the cross crown with the chiplet already mounted is the only practical approach for creating the desired profile. In the past, "kiss-lap" processes were applied in HDD head fabrication whenever an adaptation of the mechanical head interface to the disc contour was found necessary [2]. Therefore, a "kiss-lap" is executed, in our case as a nanogrinding process [3]. Such a process requires an assembled head, i.e. the slider mounted to a flexure. Furthermore, in this case, the SLIM

head has to have actuation capabilities, since it was necessary to rotate the chiplet during the process of creating the cross crown.

For positioning the SLIM head on the nanogrinding plate, a head mounting tool is necessary. The SLIM head mounting height agrees with the mounting height ("z-height") later found in the HDD. Furthermore, the head mounting tool allows to lift up the head from the plate and lower it to the plate, respectively.

For executing the process, the head whose chiplet is to be contoured is inserted in the head mounting tool, which positions it on a newly conditioned and prepared nanogrinding plate. The plate starts rotating and the head lowered to the plate. By appropriately rocking the chiplet through microactuator excitation, the chiplet is profiled until the desired cross crown is reached.

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FLEXIBLE POLISHING WITH BOUNDED GRAINS FOR COMPLEX CERAMIC ENDOPROSTHESES

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INTRODUCTION

Worldwide, the knee is the most medically treated joint. The complication rate amounts to about 26% on the whole. The main reason for the failure of the interconnection between bone and prosthesis is the agglomeration of wear debris of the polyethylene part [1].

Inspired by the successfully employed ceramic hip joint prostheses, the trend in the development of implant material pairings goes towards the use of ceramics [2-3]. So the objective of this project within the scope of the Collaborative Research Center 599 sponsored by the DFG is to improve the durability of complex implants, like knee implants, by means of new machining technologies for ceramics [4].

The objective is to realize automated high-precision machining processes using one single machine, so that the low-wear ceramic also gets the upper hand for other complex joint replacements. Two promising polishing methods using flexible tools with bound abrasives are presented. On the one hand, rubber bond diamond tools are applied with special focus on the modeling of the influencing parameters. On the other hand, a polishing belt apparatus is introduced including first experimental results.

MATERIAL REMOVAL MECHANISMS

In general, ceramic materials are machined using abrasive processes with diamond particles. For controlling the surface properties, the involved material removal mechanisms have to be characterized. In the most commonly applied machining processes with geometrically non-defined cutting edges like grinding, honing, lapping and polishing, the material is removed by the penetration of separate diamond cutting edges into the working material.

Zum Gahr [5] classified the mechanisms involved in the wear of the material. Two of these mechanisms are relevant for the polishing of ceramics using flexible tools with bound grains (FIG. 1). The penetration of the grains mostly causes plasticizing and a ductile removal of ceramic particles, but unbound grains, which are rolling between the flexible bonding and the

workpiece, can cause micro-cracks and brittle outbursts.

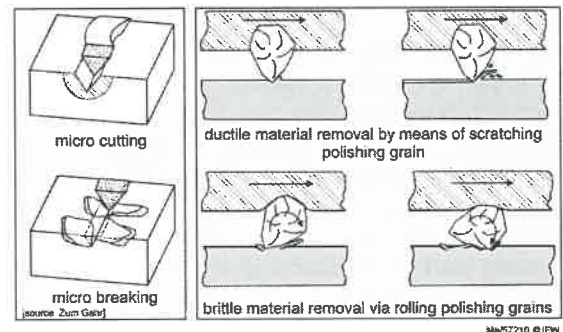


FIGURE 1. Polishing mechanisms.

The superposition of different scratching motions of the abrasive grains parallel to the work piece surface is characteristic for abrasive processes. Taking this into consideration, a hypothesis can be put forward for polishing processes using flexible tools with bound grains (FIG. 2).

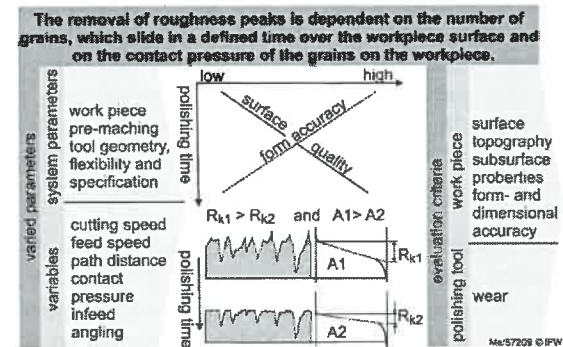


FIGURE 2. Hypothesis and relevant parameters.

The removal of the roughness peaks depends on the number of grains which slide over the work piece in a fixed period of time and on the contact pressure of the grains on the work piece. A higher number of grains leads to decreased surface roughness and a higher contact pressure of the polishing grains on the work piece leads to an increased smoothing up to a certain degree. If the pressure is increased beyond this certain point, the material removal rate also increases. This means that not only the surface roughness, but also the shape accuracy are

reduced with an increase in the contact pressure or the polishing time.

In the left of FIG. 2, the relevant actuating variables and system variables are summed up; the evaluation criteria are listed in the right. The modeling procedure presented in the following will point out the relationship between the influencing variables and the evaluation variables. The knowledge of this relationship is necessary to predict of the work piece quality.

MODELING

The first focus in the modeling process is on the number of polishing grains which slide over one unit of the work piece.

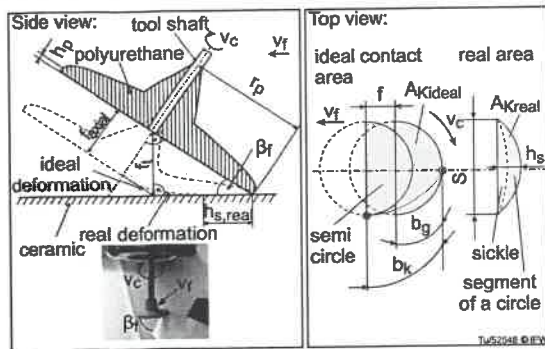


FIGURE 3. Variables of a flexible polishing tool.

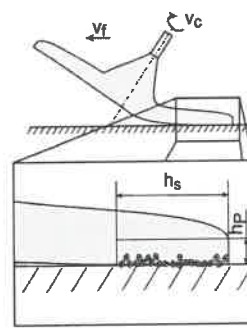
With an infeed depth f_t of the flexible polishing tool into the work piece, a specific forming of the tool is achieved by means of the contact pressure p_A depending on the feed angle β_f (FIG. 3). If we assume that the tool is ideally elastic, the tool contour is linear up to the point where the tool bends at the edge of the contact area. However in combination with the softness resp. rigidity c_p of the tool, the real deformation differs from the ideal one. The work piece contour is nonlinear from the middle of the tool and passes tangentially into the contact area. This also affects the ideal resp. real contact areas A_{Kideal} and A_{Kreal} , which have to be defined by experiments. The number of polishing grains can be determined as follows: First, the number of grains per unit of volume N'''_{Kp} is defined for the concentration C , the single grain volume V_{ge} and the density of the polishing material ρ_g ,

$$N'''_{Kp} = \frac{C}{(V_{ge} \cdot \rho_g)} \quad (1)$$

whereas $\rho_g = 3,52 \frac{g}{cm^3}$ for diamond material

and $V_{ge} = q_e \cdot \frac{1}{6} \cdot \pi \cdot d_g^3$ (2)

whereas $q_e = 1$ the grain shape factor for round grains.



Assumptions:

- ♦ contact area is circle segment
- ♦ uniform diamond distribution in the tool
- ♦ unused tool

Correlation:

$$N_{Kp}'' = \frac{3 \cdot C \cdot h_p \cdot s \cdot (r_p - h_s)}{\pi \cdot \rho_g \cdot q_e \cdot d_g^4}$$

with:

- N_{Kp}'' no. diamond grains in overlaid tool area
- C concentration
- ρ_g density of diamond grain
- q_e factor for the grain form
- d_g diamond grain size

TuS2548 © IFW

FIGURE 4. Grains in the contact area.

The number of grains in the overlying section of the disk $N'''_{Kp,ges}$ (FIG. 4) is

$$N'''_{Kp,ges} = N'''_{Kp} \cdot V_{p,Abschnitt} \quad (3)$$

with a polishing tool volume in the overlying section $V_{p,section}$ of:

$$V_{p,Abschnitt} = \frac{1}{2} \cdot h_p \cdot s \cdot (r_p - h_s) \quad (4)$$

whereas h_p = height of the polishing tool, h_s = height of the circle segment, r_p = radius of the polishing tool and s = length of the circle segment chord (Fig. 3 and 4). The number of grains in the overlying area N'''_{Kp} is calculated by dividing $N'''_{Kp,ges}$ by the grain size d_g :

$$N''_{Kp} = \frac{N'''_{Kp,ges}}{d_g} \quad (5)$$

Taking into account the cutting speed v_{cp} , the number of grains sliding over the contact area $N''_{Kp,v}$ is

$$N''_{Kp,v} = N''_{Kp} \cdot \frac{v_{cp}}{\pi \cdot d_p} \quad (6)$$

whereas d_p = diameter of the polishing tool.

The transformation with regard to the work piece is carried out by regarding one surface element and the grains sliding over it. Because the tool is symmetric, only one half is considered:

$$\frac{1}{2} \cdot A_K = \frac{1}{2} \cdot \frac{N''_{Kp,v}}{N''_{WE}} \quad (7)$$

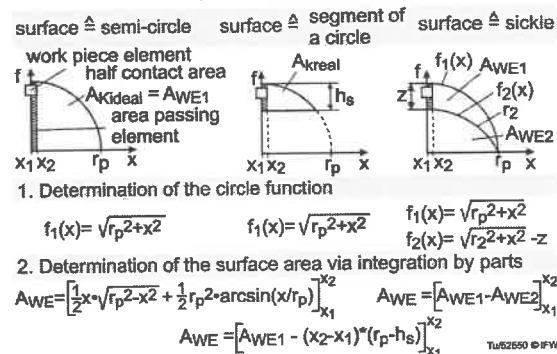


FIGURE 5. Grains per work piece element.

The ratio of the half contact area A_K to the area of the work piece element $A_{WE,i}$ is similar to the ratio of half the number of grains sliding over the surface $N''_{Kp,v}$ to the number of grains sliding over one work piece element N''_{WE} . $A_{WE,i}$ is the segment of the tool which is in contact with the work piece (FIG. 5).

The surface area of this segment can be determined using integration by parts. Three assumptions are made regarding the deformation of the contact area: in case of an ideal deformation, the area is a semi circle with the radius of the polishing tool r_p ; the real deformation corresponds to a segment of a circle with a segment height h_s or to a sickle.

For the semi-circular contact area (FIG. 5, left) follows

$$A_{WE} = \left[\frac{x}{2} \cdot \sqrt{(r_p^2 - x^2)} + \frac{r_p^2}{2} \cdot \arcsin\left(\frac{x}{r_p}\right) \right]_{x_1}^{x_2} \quad (8)$$

For the contact area in the form of a circle segment (FIG. 5, center), A_{WE} has to be calculated by subtraction of the hatched rectangular area:

$$A_{WE} = [A_{WE1} - (x_1 - x_2) \cdot (r_p - h_s)]_{x_1}^{x_2} \quad (9)$$

In the case of the sickle-shaped contact area (FIG.5, right), A_{WU1} is calculated as well as the surface area of a circular function which is translated by z and if necessary provided with an increased radius r_2 :

$$A_{WE} = [A_{WE1} - A_{WE2}]_{x_1}^{x_2} \quad (10)$$

For the calculation of the grains sliding over the work piece N''_{WE} , $A_{WE,i}$ has to be inserted into Equation 7.

The feed speed v_{fp} also influences the number of grains sliding over a work piece unit of 1 mm^2 within a fixed period of time. This has to be taken into account by means of the following calculation:

$$N''_{WE,v} = N''_{WE} \cdot \frac{v_{cp} \cdot (s + 1 \text{ mm})}{2 \cdot \pi \cdot r_p \cdot v_{fp}} \quad (11)$$

Whereas $s = r_p$ for an ideal contact area resp. $s = h_s$ for a real contact area or $s = z$ for a sickle-shaped contact area.

If the respective unit of the work piece is not located in the middle of the tool, the number of grains increases in the form of a circular function with a shifting of the unit towards the tool edge. The integration interval has to be adapted accordingly.

The modelling approach presented in this paper relates several variables to each other by means of their interdependencies. Thus the number of grains sliding over one work piece unit within a

fixed period of time can be calculated. The assumptions which have been made for this calculation have been proven by means of experiments in which the numbers of grains and the contact pressure on the grains have been varied systematically.

USE OF FLEXIBLE POLISHING TOOLS

In order to verify the presented modeling approach, the machining parameters in the polishing process cutting speed v_{cp} , feed speed v_{fp} , path distance f_p and infeed a_{ep} have been varied and the effects of these variations have been evaluated (FIG. 6).

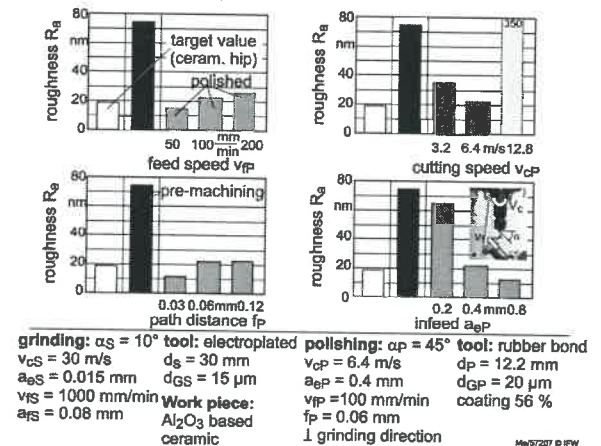


FIGURE 6. Roughness values after machining by flexible polishing tools.

The roughness values R_a attained by the polishing process are contrasted with those of the initial ground surface with $R_a = 75 \text{ nm}$ and with the surface of conventional ceramic hip prostheses with $R_a = 20 \text{ nm}$. With an increase in the feed speed v_{fp} , the surface roughness rises because less polishing grains are sliding over the surface per time unit. With a lower path distance f_p , i. e. in case of an increased overlapping, the surface quality improves because more polishing grains are sliding over each surface unit. The roughness can also be reduced by an increase in the infeed a_{ep} because the contact area is enlarged and more grains are implied. It can be assumed that additionally, the force of each grain is enhanced, which again increases the smoothing effect. In regard to the cutting speed, the optimal results are attained using a moderate speed of $v_{cp} = 6.4 \text{ m/s}$. For a target value of $R_a = 20 \text{ nm}$, medium values are also sufficient for the feed speed v_{fp} , the path distance f_p and the infeed a_{ep} .

Our hypothesis mainly complies with the experimental results. The normal force per grain is integrated into the presented model.

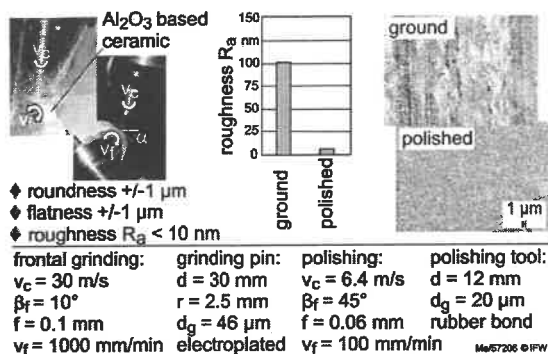


FIGURE 7. Results of the wear tests.

Several specimens have been subject to wear tests using the parameters and tool specifications presented in the bottom of FIG. 7. These specimens have been pre-machined by a front grinding process using a toric mounted point in order to attain high-quality components with $R_a < 10 \text{ nm}$ after the polishing process. The surfaces are also shown in SEM micrographs in the right of FIG. 7. The shape accuracy lies within the target tolerances of $\pm 1 \mu\text{m}$.

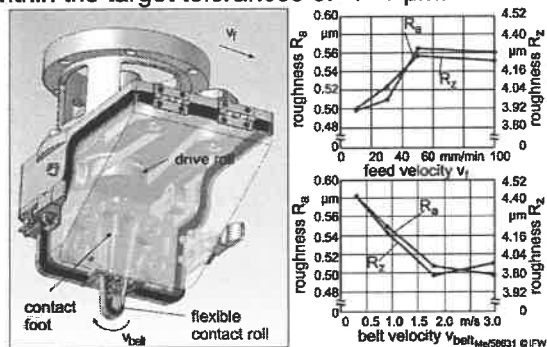


FIGURE 8. Belt apparatus – first results.

For the testing of an alternative polishing method, a belt polishing system was developed (FIG. 8). Thanks to a flexible contact roll with a rubber surface and a flexible contact foot fixed by a spring, the machining process will not be purely force-controlled. Additionally, a long belt path was implemented using eight deflection pulleys providing a higher wear volume compared with that of the rubber bond flexible polishers. A standard diamond belt with a grain size of $d_g = 25 \mu\text{m}$ was tested first. The cutting speed corresponding to the belt velocity and feed velocity were varied. The plane bioceramic specimens were pre-machined by peripheral grinding with an initial average roughness of $R_a = 0.6 \mu\text{m}$. In analogy to the results attained with the rubber bond tools, the roughness values were decreased by lower feed speeds and higher cutting speeds. The mean surface roughness R_a is developing similar to the roughness depth R_z . With

a slower feed speed and a higher belt speed, more grains are penetrating into the ceramic surface. If more grains are engaged, more work piece material is removed and the surface is smoothed further.

OUTLOOK

In the present investigations, the rigidity, the corresponding real contact area and the influence of the forces per grain on the surface quality have been determined [6, 7]. The next step will be to validate the model for flexible polishing tools by means of experiments. At the same time, the belt apparatus will be examined further regarding the optimal belt specifications and contact rolls.

Acknowledgement

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DYNAMIC DESIGN FOR IMPROVED CONTROL

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Introduction

In the design of Mechatronic Systems for Precision Positioning the design team must consider many different aspects. Multi-disciplinary trade-off is often required and each discipline can influence the system performance. In the conceptual stage of development the following four steps can be identified,

1. Assessment of the customer requirements to translate them to accuracy targets
2. Identification of disturbances and quantitative estimation of their magnitude and frequency content
3. Determination of the required feedback controller power to handle disturbances and achieve the required accuracy
4. Design the system elements, using the multi-disciplinary knowledge, such that factors limiting the controller performance are avoided

Limitations may be present in all disciplines. Here the attention will be put on designing the dynamics of the mechanical modules so as to allow for higher performance of the feedback system. This abstract first introduces the basic module that is discussed. The dynamics that influence the Controller design for the original system will be described. Next a number of steps that can be made to improve the system dynamics will be described.

SYSTEM DESCRIPTION

As an example of a mechanical module a wafer stage for a semiconductor lithography machine will be used. An early example of such a module, (ASML, 1985), is shown in figure 1 and a schematic of the system in figure 2. The wafer is supported on a mirror block that is used for interferometric measurement for the XY motion.

For vertical motion an actuator assembly supports this mirror. For motion in the X-direction a linear motor is used and the total system is supported by an air-bearing foot. The air-bearing floats over a granite base and allows for XY horizontal motion.

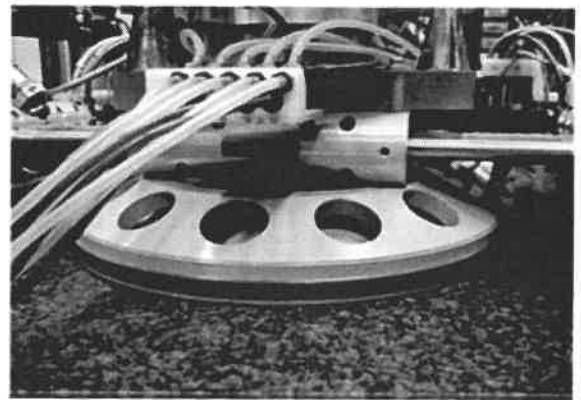


FIGURE 1. Photograph of a linear motor driven and air-bearing supported wafer stage.

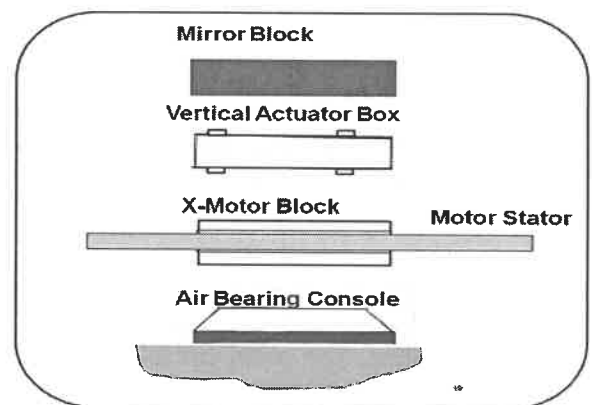


FIGURE 2. The wafer stage is build up out of four functional modules.

This system must position the wafer over a stroke of about 250 mm with position stability

better than 50 nm. Disturbances from floor vibrations and reaction forces in the frame call for a higher feedback control performance for next generation systems.

MECHANICAL LIMITATIONS

When trying to improve the gains in the controller it proved that internal dynamics were leading to system instability. To describe the system dynamics a basic model can be made. The actual analysis was done using the software tool 20SIM, produced by Controllab in The Netherlands. A picture of the iconic input to the simulation is shown in the appendix. Here the system is modeled with two bodies, movable in the XZ plane. One body combines the Motor and Air-foot. The other body combines the Mirror with the Actuator box.

Using this model the Open Loop Frequency Transfer function for the mechanical system can be determined. This is the transfer from a driving force applied to the motor to the measured displacement at the interferometer. This Transfer Function is shown in figure 3.

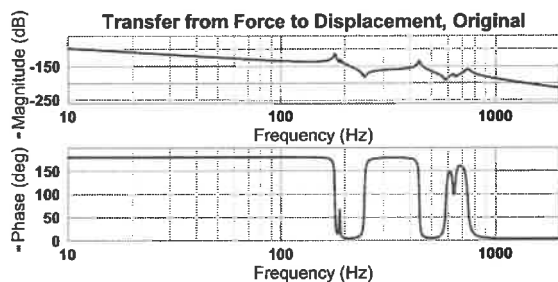


FIGURE 3. In the transfer function a clear resonance around 180 Hz can be seen.

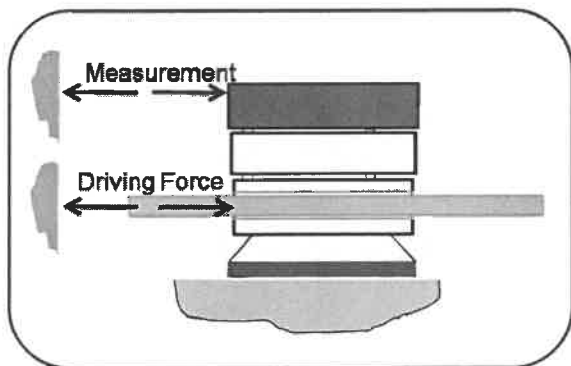


FIGURE 4. From this picture it is clear that the driving force is applied below the Center of Gravity. The Measurement is done above this CoG.

In the Transfer Function the presence of a resonance around 180 Hz is clearly visible. A serious negative phase shift of 180 degrees is due to this resonance.

When one observes the schematic drawing of the system in figure 4 it is clear that the driving force is applied below the Centre of Gravity. As a result the application of the Force will lead to a moment on the air-bearing trying to prevent rotation of the body. At 180 Hz the bearing and the masses and inertia of the stage lead to a resonance behavior. In the motion associated with hat resonance the Mirror will move in opposite direction compared to the point where the force is applied. This explains the origin of the resonance seen in the Transfer Function.

As this type of effect is common in many such systems a good guiding principle during design in a linear motor system is;

The driving Force should be applied through the Centre of Gravity and should be directed exactly in the desired direction of motion.

IMPROVED DESIGN

In our example system one can make a redesign to achieve this. In figure 5 a schematic drawing is shown. The module with the linear motor is moved up and the Actuator box is moved down. This will make it possible that the driving force acts through the Centre of Gravity.

This design leads to additional complexity in the mechanical construction as the Actuator Box should be rigidly connected to the Mirror and the Motor must be fixed to the Airfoot. Still the added mechanical complexity is offset by the potentially better performance.

When the internal stiffnesses in the system are infinitely large the ideal Transfer Function is obtained. The transfer function will be equal to that of one mass freely moving in one direction.

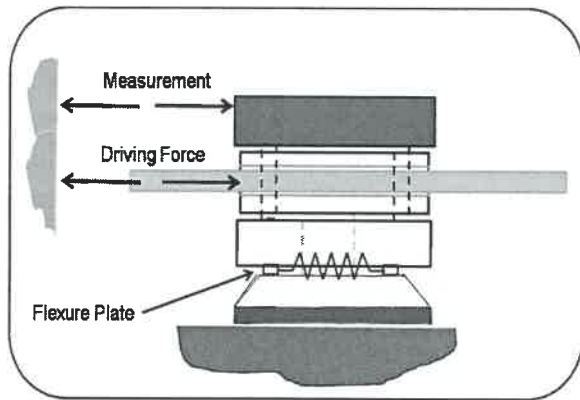


FIGURE 5. The Flexure Plate connects the two parts in the horizontal directions.

The real system however will have internal stiffness between the two elements. In the vertical direction the top module, Mirror and Actuator Box, will rest on the Airfoot with a limited stiffness. In the horizontal direction the top and bottom module will also be connected with a connector allowing for relative vertical motion. For this a flexure plate can be designed in the fixation between the two parts. In figure 5 is shown where this flexure plate is placed.

For this new system the Transfer Function, from driving force to measured displacement is calculated. The result is shown in figure 6.

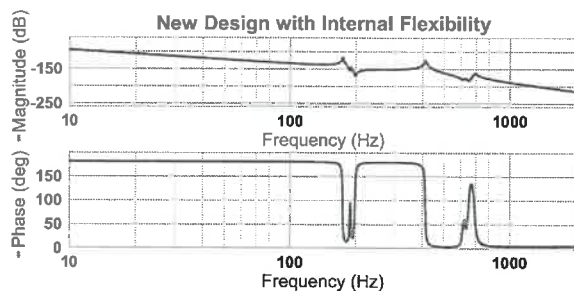


FIGURE 6. The Transfer Function for the Improved system still contains a significant resonance at 200 Hz.

From this graph it is clear that the "improved" system still shows a significant resonance around 200 Hz. Although the module is driven in the overall Centre of Gravity resonances due to the limited stiffness of the air-bearing will still be limiting the system control performance.

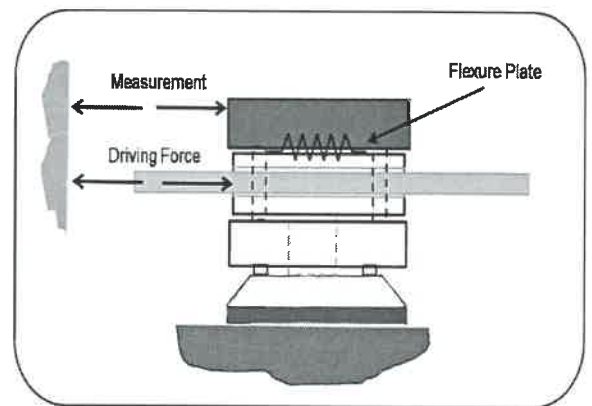


FIGURE 7. The Flexure Plate location is changed to reduce the internal vertical loading of connections and air-bearing.

SECOND IMPROVEMENT STEP

The excitation of the resonances in the air-bearing must be due to vertical forces acting internally in the design. When the two rigid bodies are considered separately it becomes clear what is happening.

The driving force acts at the Centre of Gravity for the total module. On the bottom module the horizontal force from the flexure plate, needed to accelerate the upper mass, is also acting. Those two forces combined will create a moment around the CoG of the bottom part, in turn leading to excitation of rotations.

To eliminate the occurrence of this moment the driving force and the force needed to accelerate the upper mass might be manipulated such as to lead to a net zero moment around the CoG of the bottom part. Given the two bodies and the driving force acting through the total CoG it turns out that the force needed to accelerate the upper mass must be applied through the CoG of the upper mass. This can be achieved by mounting of the flexure plate in the right plane. In figure 7 this new design is shown.

The resulting transfer function for the final design is shown in figure 8.

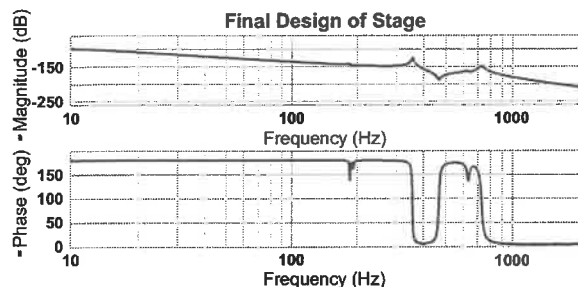


FIGURE 8. For the new design the Transfer Function indicates that the resonance around 180 Hz is almost complete removed.

CONCLUSION

In this paper it is shown how design rules can be used to create better designs. The initial change to drive through the Centre of Gravity proved to be insufficient to obtain a serious improvement. Further application of the same principle for the two separate parts in the design leads to significant improvement that will allow the controller to have better capability to handle disturbances.

APPENDIX

In the model shown in figure 9 the schematic model used for the simulations is shown. There are two masses that can move in two directions, X and Z, and make small rotations R_y .

In the upper right side the sensor measuring the motion of the upper mass at the desired point is shown. On the left the actuator driving the lower mass is shown.

Vertical and horizontal springs and dampers are connecting the two bodies and are used to model the air-bearing.

The model is made using the iconic modeling in the 20SIM software package distributed by Controllab Products BV in the Netherlands.

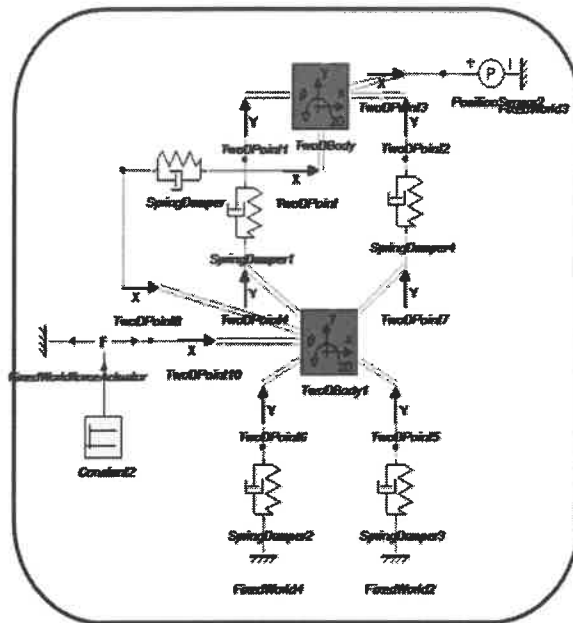


FIGURE 9. For the new design the Transfer Function indicates that the resonance around 180 Hz is almost complete removed.

MR FLUID ACTIVATED SPHERICAL JOINT FOR PRECISION ORIENTATION CONTROL

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SYNOPSIS

In this paper, we present the design, modeling, optimization and preliminary characterization of a spherical ball joint whose orientation may be locked, and unlocked via the use of Magneto-Rheological (MR) fluids. This machine element was investigated to ascertain utility in precision mechanisms, e.g. locking joint for mechanisms that steer optics. The geometry of the ball, receiving socket and the MR-filled gap between them has been optimized to maximize the locking torque per unit power. This is accomplished by distributing the magnetic flux density throughout the MR fluid. The joint is designed with a permanent magnet to enable passive locking of joint orientation in a non-energized state. This enables long-term position locking without power dissipation and subsequent thermal errors. The joint is energized by routing current through a coil that partially cancels the field from the permanent magnet. This unlocks the MR fluid and makes it possible for an actuator to position the joint.

INTRODUCTION

Surgical stands facilitate operations, by enabling a surgeon to dynamically position, move, and lock tools throughout a procedure [1]. These stands are manually positioned by the doctor and then manually locked into position, usually via a cumbersome cable system [2]. The combination of stick-slip in the stand's joints and cumulative Abbe error between them makes it difficult to rapidly and accurately set position/orientation of surgical tools. There are also numerous applications in macro- and meso-scale optics wherein precision orientation and long-term fixation, e.g. optics, is important. Precision spherical joints have been studied before for many applications because of their high stiffness and accuracy [3].

Herein we discuss the use of MR fluids to create a precision ball-socket joint. MR fluids' magnetic field dependant shear stress allows for locking to

occur without any mechanical parts and can therefore be used to create a fine resolution joint. Figure 1 shows a CAD model of the joint.

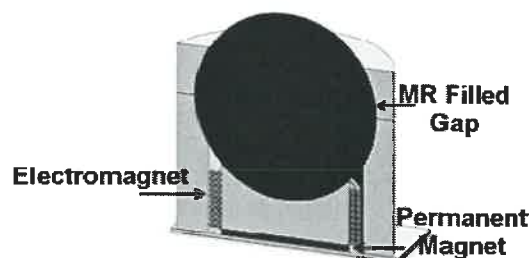


FIGURE 1. Cross-section of MR spherical joint.

The joint's outer housing, comprised of steel, encloses a steel ball. An MR fluid, (impregnated with of 200 micron diameter glass spheres that make up approximately 5% of the fluid volume) fills the gap between the ball and housing. The glass spheres serve to maintain the gap between the housing and ball. The ball and housing act to guide the field lines within the joint as shown in Fig. 2.

Within active fluids, MR fluids are categorized as 'positive gain,' which means that their viscosity increases as a function of the magnetic field. Past designs have sought to induce a locking via actively powering a field, thereby requiring sustained, high power inputs to maintain locking. The power is often dissipated as heat via joule heating within the coils that induce the field. The heating yields thermal growth errors that preclude the use of these devices in many precision applications. The main point of this paper is to show that a suitable combination of coil and permanent magnet may be used to create a joint with a fixed "off" state and low power requirements for intermittent unlocking.

The MR fluid is passively activated by the field of the permanent magnet, thus locking the joint with no power consumption. An electromagnet is used to counteract and reroute the field, thereby deactivating the fluid. The counteracting field

reduces the shear stress in the MR fluid enough to permit motion of the joint when it must be reoriented. This joint is "locked" when powered off and powered to "unlock" during the short times required to move the joint.

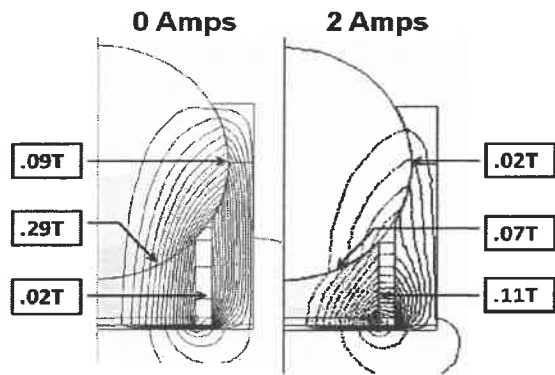


FIGURE 2. FEMM model of the MR joint, showing the magnetic field at 0 and 2 Amps.

CHARACTERISTICS OF MR FLUIDS

In an MR fluid, the viscosity of the fluid resists motion via shear stress across the gap between the socket and ball. When the magnetic field is applied, the iron particles align in chain-like formations; these restrict the fluid movement and increase the resistive shear stress. Common MR fluids exhibit dynamic yield strengths from 1kPa to 50 kPa for applied magnetic fields of 150-250 kA/m [4]. Their yield strength is proportional to the applied magnetic field. As a result, MR fluids may exhibit up to a 50:1 increase in shear stress with a response time on the scale of milliseconds [5].

Our MR fluid consists of iron particles in a hydrocarbon. The fluid, Lord MRF-132DG, contains 1 to 20 micron sized particles. Figure 3 shows shear stress as a function of field, extracted from technical data provided by Lord.

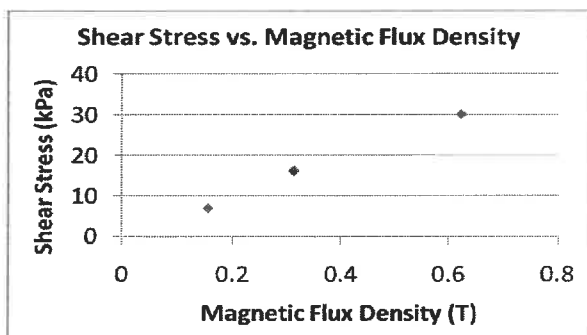


FIGURE 3. Relationship between stress and magnetic flux density for Lord MRF-132DG.

This graph covers the flux densities that are relevant to our application. For flux densities larger than 0.7 Tesla, the fluid begins to saturate and the shear stress plateaus.

MODELING AND OPTIMIZATION

When an MR fluid is used in shear mode the important parameters are surface area and magnetic field across the MR filled gap. As such, a main goal of the design is to assure the distribution of magnetic field between the ball-socket contact is large in magnitude and uniform in distribution. This ensures that the MR fluid is activated more evenly and strongly, which in turn yields the highest resistive stress over the surface of the ball-socket contact. A second goal of the design was to minimize the power required to unlock the joint. Others have attempted to unlock MR joints solely via cancellation of the field from permanent magnets. Our design makes use of cancellation, but also partially reroutes the magnetic flux away from the MR-filled gap when the electromagnet is activated. This leads to additional reduction of resistive torque per unit of activation power.

The joint's resistive torque may be calculated via Eq. 1, where the shear stress, σ , was obtained by estimating the shear stress equivalent to the magnetic field given by a Finite Element Magnetism Method (FEMM) simulation.

$$\tau = \sigma r^3 \int_0^{2\pi} \int_{\theta_{min}}^{\pi} \sqrt{\cos^2(\theta) + \sin^2(\theta) \cos^2(\phi)} \sin(\theta) d\theta d\phi \quad (1)$$

The radius of the joint sphere is represented by r , and θ_{min} represents the half angle of the sphere that is not covered by the socket. Eq. 1 may be rewritten as Eq. 2, where C is a non-dimensional constant that is associated with the torque producing surface for a given value of θ_{min} .

$$\tau = C \sigma r^3 \quad (2)$$

JOINT DESIGN AND MODELING

Achieving Negative Gain

The concept of negative gain in MR-based devices has been achieved by Lord Corporation

in the design of MR valves. The valves used a permanent magnet to activate a fluid and an electromagnet to cancel and redirect the magnetic field away from the MR filled gap [6]. Our design uses similar strategies for cancellation and redirection.

Magnetic Field Distribution and Optimization

The resistive torque depends upon the MR fluid's resistive stress, which in turn depends upon the magnetic field. Optimizing the distribution of the field across the gap in the on and off states is therefore important to maximizing and minimizing the resistive torque. FEMM modeling was used to optimize the field which the permanent magnet induced in the gap (0 Amp case in Fig. 2) and the field which the permanent and electromagnets induced (2 Amp case in Fig. 2).

The results in Figure 2 warrant explanation before moving on. When the electromagnet is turned off, the resistance of the main circuit is low given the high magnetic permeability of steel and the sub-millimeter gaps. This yields a large flux in the magnetic circuit. When the electromagnet is energized, its field reduces the field intensity over the main circuit, reducing the amount of flux passing through this path. The reduced flux draw on the permanent magnet increases the field intensity across the 'short circuit' path around the side of the magnet, drawing more flux through the large air gap path. Figure 2 shows the flux 'shorting' across the larger air gap and the corresponding decrease in field through the gap.

EXPERIMENTAL RESULTS

A prototype joint was created with a N42 grade rare earth magnet. The remaining structure was constructed from 12L14 Carbon steel, and the electromagnet was fabricated by coiling 27 gauge copper wire around a 19mm diameter steel cylinder. The components of the prototype joint are shown in Figure 4 and their salient characteristics are shown in Table 1. The gap between the socket and the ball was nominally set at 200 microns so that at least 10 MR particles could fit across the gap and enable the formation of MR particle chains. The MR fluid was mixed with glass beads that are on average 200 microns in diameter at a volume fraction of 5 %. This helped to maintain the gap.

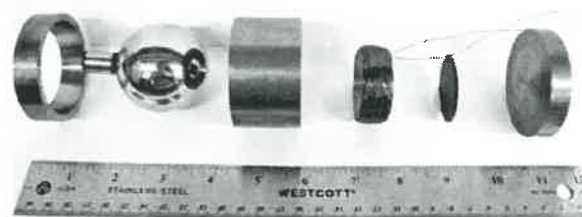


FIGURE 4. Exploded view of the prototype joint.

TABLE 1. Key dimensions of prototype joint.

Parameter	Dimension
Ball Diameter	50.8 mm
Magnet Diameter	38.1 mm
Magnet Thickness	1.6 mm
Electromagnet Winding	~270 turns
MR filled Gap	0.2 mm

Holding Torque

A force was applied to the base of the shaft that protrudes from the ball. A Wagner Force One FDIX 5 sensor was used to measure the magnitude of the force that was required to overcome the resistive a force. The sensor has an accuracy of 0.02N. The breaking force was measured for a variety of coil currents and converted to breaking torque and compared, in Figure 5, with the predictions of Equation 2.

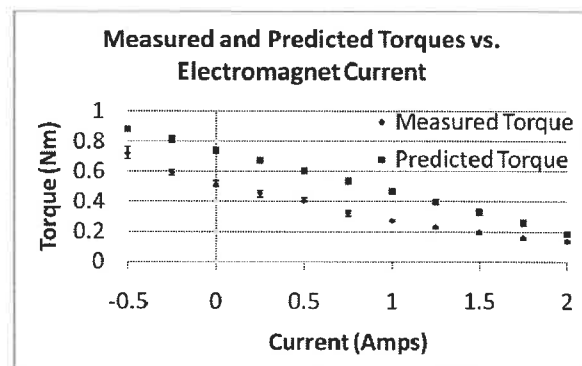


FIGURE 5. Comparison of measured and predicted breaking torques. Error bars represent maximum and minimum measured values.

As Figure 5 shows, the holding torque of the joint may be controlled by varying the amount of current applied to the electromagnet. The measured torque was only 60-80% of the predicted torque. The two most significant sources of error are: that in the prototype there was a significant air gap between the two steel shells. This air gap increases the resistance of

the main flux path, which decreases the magnetic flux through the MR filled gap. Therefore model will predict a higher shear stress. The other significant source of error is the limited data from where the magnetic flux to shear stress relationship was obtained. As a result the shear stress for a predicted magnetic flux is a very rough estimate.

Creep Test

Figure 6 shows the setup for the creep test. The stand was constructed from aluminum to ensure that the magnetic field within the joint would not be affected. Three cap probes were positioned so as to record (i) the displacement of the shaft that is attached to the ball and (ii) the displacement and rotation of the base.

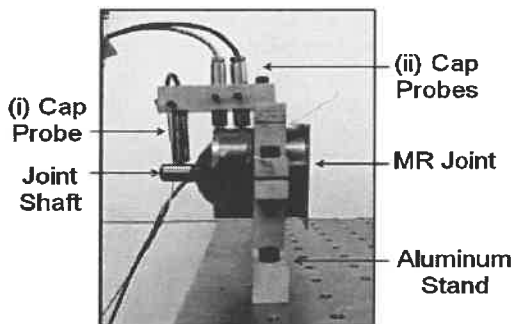


FIGURE 6. Setup used to measure creep.

During the test, a 100g load was hung from the base of the shaft. We anticipated several microns of motion during a four hour test and therefore the capacitance probes were used in the coarse setting. The resolution of the probes on this setting was 40nm. The relative motion of the shaft and socket joint were measured over four hours.

The relative displacement is shown in Fig. 7. There is a steady change in relative position of approximately $25^{nm}/hr$, which we attribute to the creep of the joint. The average trend of the displacement is denoted by the black line in the middle of the data. The outer bounds of the capacitance probe resolution are denoted by the two gray lines that are parallel to the average trend line. The higher order frequency seen in the data is due to the air exchanges in the room from the HVAC system.

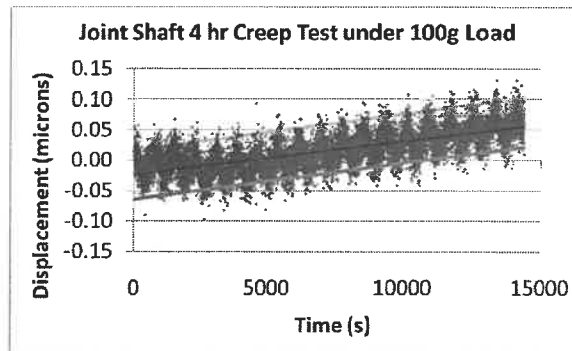


FIGURE 7. Shaft creep over a 4 hour period.

CONCLUSION

The preliminary characterization of the design suggests that it will be suitable for use in several types of locking-joints that are needed in orientation mechanisms, e.g. robotics, optics and surgical stands/tools. One could imagine several robots that need a joint with current controlled torque, as well as orthopedic braces whose resistance can easily be adjusted if wear of the contacting surfaces may be mitigated.

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MAGNETIC LEVITATION FOR PRECISION MOTION WITH ONLY TWO OFF THE SHELF LINEAR MOTORS AND A NOVEL PASSIVE MAGNETIC BEARING DESIGN: MODELING AND CONTROL CHALLENGES

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1. INTRODUCTION

The development of new type magnetic materials has made a Passive Magnetic Bearing (PMB) an attractive option in the bearing industry. Its application can be found in turbo machinery, storage flywheels, high-speed turbo compressors, or in space instrument applications [1]. The advantages of PMB over other types of bearings (roller bearing, air bearing...) are its contactless operation, potential for miniaturization, simplicity and cost effectiveness [2].

Due to Earnshaw's theorem [3], the PMB system is unstable by nature. Therefore actuators or another type of bearing are required to stabilize the system. In this paper, we will present a novel design of a linear PMB in combination with two off the shelf linear motors. Also the modeling and control of this system is carried out. Finally experimental results are presented to confirm the effectiveness of the proposed design.

2. EXPERIMENT SETUP

A technical drawing of the setup is presented in figure 1.

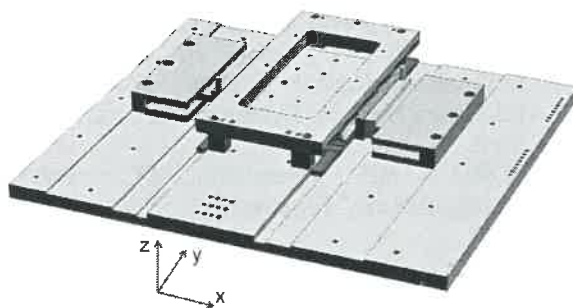


FIGURE 1. Magnetic levitation platform.

The setup consists of a bearing part and an actuation part. The actuation part consists of two off the shelf linear motors with NForcer Technology [4], which means that one motor can simultaneously provide two forces along two perpendicular axes independently. Two motors are attached at two sides of the levitated slider to control three degrees of freedom (two translations and one rotation) in the horizontal plane.

The bearing part is described in figure 2. It consists of two parallel pairs of magnet tracks attached to the stationary part and eight magnets attached at the corners of the levitated slider. The magnets attached to the slider have opposite polarization with respect to the stationary magnet tracks and thus are repelled by these static magnet tracks.

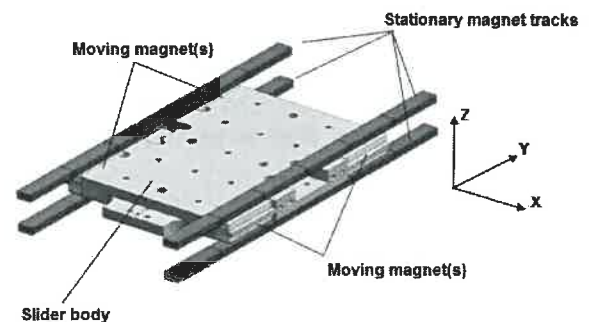


FIGURE 2. Passive magnetic bearing.

3. MAGNETIC BEARING MODELING

The bearing part of the system consists of cuboidal magnets on the moving and stationary part. To model this part, the analytical formula of the forces exerted between two cuboidal magnets given in [5] is used. The dimensions of stationary and moving magnet are given in table 1 below:

TABLE 1. Bearing parameters

Parameter	Value
Stationary and moving PM width	8 [mm]
Stationary and moving PM height	4 [mm]
Stationary PM length	180 [mm]
Moving PM length	30 [mm]
Air gap	1.8 [mm]

Based on parameters in table 1, the magnetic force in x direction is estimated. The result is plotted in figure 3.

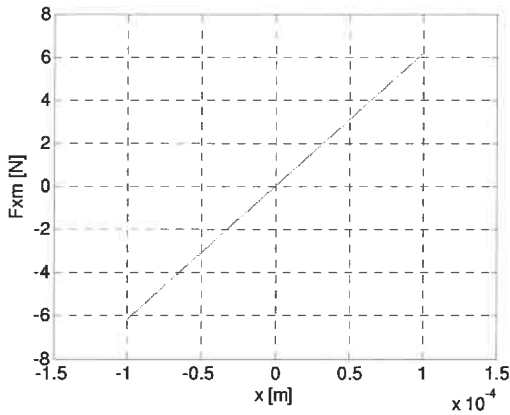


FIGURE 3. Force and displacement relation in x direction.

We see that when the displacement is small, the relation is linear and the stiffness in x direction can be approximated by:

$$k_x = -\frac{\partial F_{xm}}{\partial x} = -61.79 \cdot 10^3 \text{ [N/m]} \quad (1)$$

Similarly the force and displacement in z direction are described in figure 4.

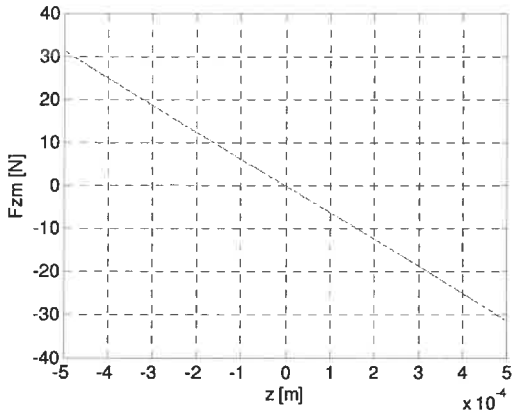


FIGURE 4. Force and displacement relation in z direction.

The stiffness in z direction is then approximated

$$\text{by } k_z = -\frac{\partial F_{zm}}{\partial z} = 62.40 \cdot 10^3 \text{ [N/m]} \quad (2)$$

The magnetic force in y direction is approximately zero because the stationary magnet track is much longer than moving magnets, hence the stiffness in this direction is approximately zero. This result also agrees with Earnshaw's theorem:

$$k_x + k_y + k_z = 0 \quad (3)$$

Finally, in order to control the rotational angle around z, the rotational stiffness k_{Rz} also needs to be estimated. When the rotational angle Rz is small, its stiffness k_{Rz} can be approximated by:

$$k_{Rz} = k_x \cdot l_{my}^2 \quad (4)$$

With:

l_{my} : Distance from center of gravity (COG) of slider to COG of moving magnet in y direction.

4. SYSTEM MODELING AND CONTROL

The equation of motion of the levitated slider:

$$m\ddot{x} = -k_x \cdot x + F_x$$

$$m\ddot{y} = F_y \quad (5)$$

$$J_z \ddot{R}_z = -k_{Rz} R_z + T_z$$

With:

m, J_z : Mass and inertia of slider.

F_x, F_y, T_z : Forces and torque acting on COG of slider in x, y, R_z direction respectively.

k_x, k_{Rz} : Negative stiffness of the bearing in x and R_z direction respectively.

The control structure for this system is given in figure 5. It consists of three linear controllers (C_x, C_y, C_{Rz}), input transformation matrix (T_u) and output transformation matrix (T_y).

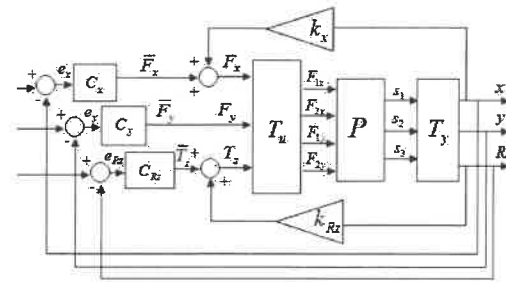


FIGURE 5. Controller structure.

Based on the bearing stiffness k_x and k_{Rz} estimated in previous section, the control law is given by:

$$\begin{aligned} F_x &= k_x \cdot x + \bar{F}_x \\ F_y &= \bar{F}_y \\ T_z &= k_{Rz} R_z + \bar{T}_z \end{aligned} \quad (6)$$

After applying the control law, the equations of motion (5) are given by:

$$\begin{aligned} m\ddot{x} &= \bar{F}_x \\ m\ddot{y} &= \bar{F}_y \\ J_z\ddot{R}_z &= \bar{T}_z \end{aligned} \quad (7)$$

Using (7), three PID controllers are designed for three Dofs x, y, Rz respectively.

The theoretical forces and torque acting on COG of slider (F_x , F_y , T_z) are transformed to four physical force inputs of two linear motors (F_{x1} , F_{y1} , F_{x2} , F_{y2}) by the input transformation matrix T_u . The three degrees of freedom of the system are monitored by a combination of two inductive sensors and one linear encoder. The three measurements are combined and transformed to the COG positions of the slider by output transformation matrix T_y .

NEGATIVE STIFFNESS CALIBRATION

The negative stiffness of the bearing is calibrated with the help of a dSPACE system. The schematic of the whole system is shown in figure 6.

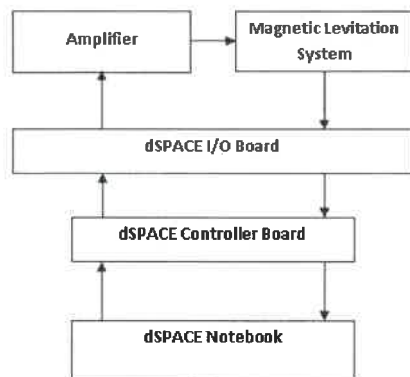


FIGURE 6. Control system block diagram

The system is positioned at different locations in x (Rz) under closed loop control, while capturing the total required controller force (torque) to keep the system at prescribed locations. This real time capturing is performed by using Mlib/mtrace library of the dSPACE system. The captured values are then used to calibrate system stiffness.

EXPERIMENTAL RESULTS

The experiments are carried out with the control system as depicted in figure 6. By using the previously mentioned controller, the system has been stabilized. The error in the standstill case in x, y direction is within 1 micrometer and the error in Rz direction is within $15 \mu rad$. These errors are the performance limits for the system due to the resolution of measurement system.

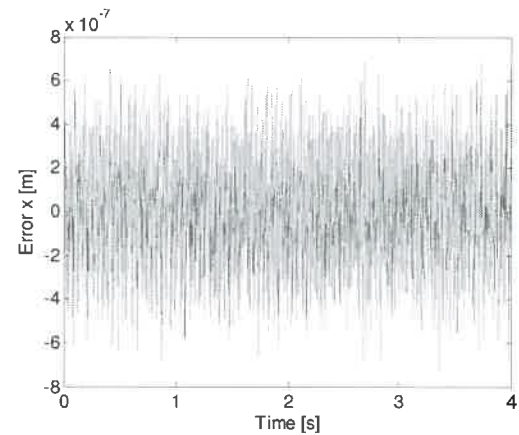


FIGURE 7. Standstill error in x direction.

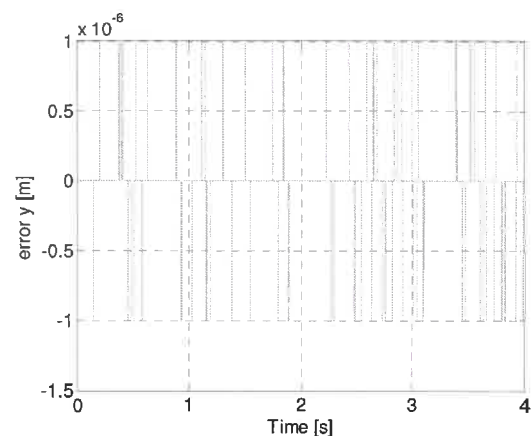


FIGURE 8. Standstill error in y direction

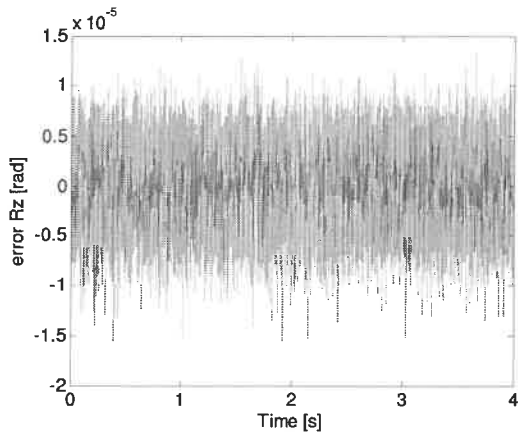


FIGURE 9. Standstill error in Rz direction.

Figures 10 to 12 show the response of the system to a third order reference trajectory. The results show that the system tracks the reference trajectory successfully.

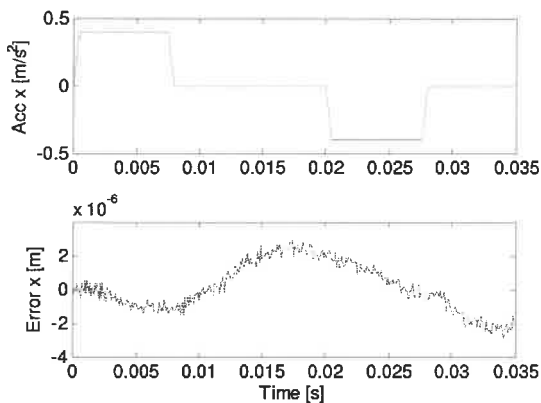


FIGURE 10. Third order trajectory response in x direction with $x_e = -6e-5[m]$, $vel = 0.003 [m/s]$, $acc = 0.4[m/s^2]$, $jerk = 800[m/s^3]$.

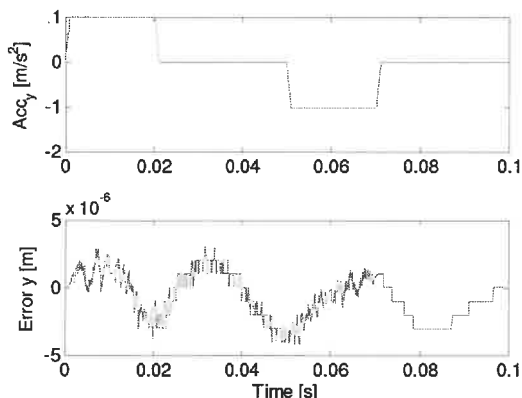


FIGURE 11. Third order trajectory response in y direction with $y_e = 0.001[m]$, $vel = 0.02[m/s]$, $acc = 1[m/s^2]$, $jerk = 1000[m/s^3]$.

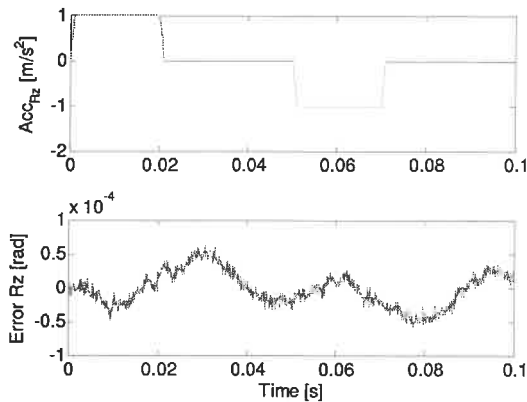


FIGURE 12. Third order trajectory response in Rz direction with $Rze = 0.001[rad]$, $vel = 0.02[m/s]$, $acc = 1[m/s^2]$, $jerk = 1000[m/s^3]$.

Conclusion

This paper presents the development of a new planar PMB system. The system is magnetically levitated by the repulsive forces of permanent magnets in three degrees of freedom. The remaining degrees of freedom are controlled by two off the shelf linear motors with NForcer Technology. The passive bearing part is modeled by using an analytical model which provides a good approximation of the bearing stiffness. The system equations and stiffness calibration have been discussed. Experimental results show that the system has been stabilized and the achieved performance is satisfactory. The system promises a low cost alternative for linear motion systems with magnetic bearings in industrial applications. In future work we will quantify and optimize the accuracy of the PMB system.

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VALIDATION OF THE PERFORMANCE OF A HIGH-ACCURACY COMPACT INTERFEROMETRIC SENSOR

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INTRODUCTION

This paper describes the metrology for the validation of displacement measurement performance of a new high-accuracy, multi-channel, fiber-based, low drift, absolute distance measuring interferometric sensor system. The system was designed for demanding applications requiring thermally passive, electrically immune, compact sensors with an extremely long mean time between failure (MTBF).

Although the system is capable of displacement and absolute distance measurement, the main focus of this paper is the metrology used to validate the latter. The performance of the system under test (SUT) is compared to a modified commercial displacement measuring interferometer (DMI) system (Zygo ZMI 4000). Uncertainty contributors in the validation metrology (VM) system are minimized through a compact metrology loop and compensation of Abbé errors. The design of the validation metrology is driven by a comprehensive uncertainty budget, with resulting displacement uncertainties ($k=2$) at the part per million (ppm) level. A detailed description of the setup and uncertainty budget is provided.

THE SYSTEM UNDER TEST (SUT)

The system under test is described in detail in these proceedings and elsewhere [1]. The system is designed to operate over a range of $\pm 250 \mu\text{m}$ at a standoff of 1.2 mm. It operates at near-IR ($\sim 1550 \text{ nm}$) wavelengths and has sub-nanometer resolution.

VALIDATION METROLOGY

The displacement measurement performance of the displacement sensor under test is validated by comparison to a modified commercial DMI. In order to effectively validate the performance of the SUT, a VM method with commensurate measurement uncertainty is required.

The Setup

A schematic of the VM setup is shown in Figure 1.

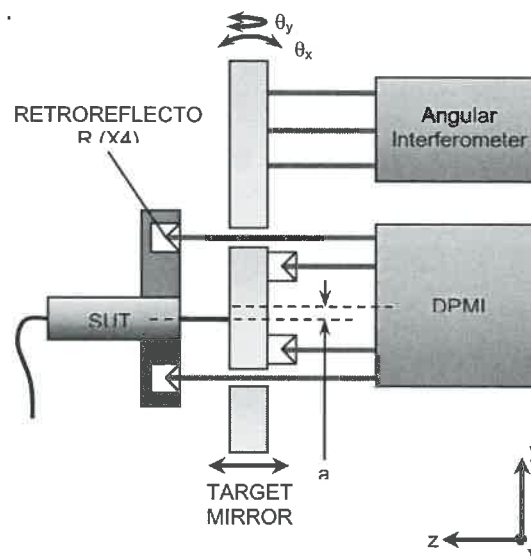


FIGURE 1. Schematic of validation metrology setup.

The heart of the system is a target mirror the displacement of which is measured by both the SUT and the VM. The VM is implemented as a modified differential plane mirror interferometer (DPMI) configuration (Zygo 6191-0188-02) that performs a differential measurement between the target mirror and the mount of the SUT. The beams that probe the mount of the SUT pass through holes in the target mirror. This configuration differs from the standard DPMI configuration in that the plane mirror target on the SUT mount has been replaced by retroreflectors to eliminate any adjustments between the SUT and its mount which could compromise stability. The retroreflectors allow the SUT orientation to be adjusted to orient it normal to the target without compromising alignment of the VM. In addition, retroreflectors on the back of the target mirror also allow for tilting the target mirror to facilitate testing of sensor performance in the presence of mirror tilt. While the figure gives the appearance that the measurement and reference beams of the DPMI are coplanar, in reality, the two beam pairs are

in two mutually orthogonal planes. The reference beams (the ones that pass through the target mirror) are in a plane normal to the plane of the paper, while the measurement beams (the ones that are reflected from the back of the target) are in the plane of the paper. This particular arrangement of beams requires the DPMI to be rotated 45 degrees (about an axis parallel to the beam axis) relative to its standard mounting configuration, resulting in a non-standard mounting configuration. Also, since this interferometer is a polarization based device, the incoming polarization states have to be rotated in order to launch correctly oriented polarization states into the interferometer. The DPMI measures the average of the change in optical path length (OPL) in the two legs of the measurement or reference beam. In other words, the effective point of measurement is midway between the beams, and must be arranged such that this point coincides with the location at which the displacement is to be measured. A second interferometer (Zygo 6191-0624-01) system capable of measuring angular motions of the target mirror is used in tandem with the DPMI to measure the angular motions (θ_x and θ_y) of the mirror. This information is used to correct for contributions due to the Abbè offset (a in Figure 1). The target mirror is coated on one side to provide a reflecting surface for the angular interferometer. The other side (closest to the SUT) is an uncoated bare glass surface that simulates the typical target for the SUT.

The Measurand

The ultimate goal of the measurement is to determine the fidelity with which the SUT measures the relative displacement between the SUT (specifically the reference surface of the SUT) and target mirror. In order to make this determination, an independent measurement of this displacement is required. The required level of performance is achieved by careful analysis of the metrology loops (see Figure 2).

While a direct measurement of the target mirror relative to reference surface of the VM would provide the ideal metrology, practical considerations result in a more indirect measurement, i.e., between retroreflectors mounted in the mount for the SUT and the rear surface of the target mirror.

The advantages of the differential nature of the measurement may best be understood by

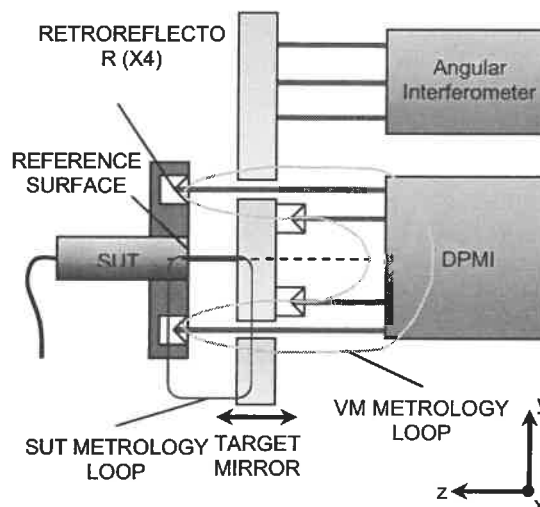


FIGURE 2. Metrology loops for VM and SUT.

contrasting this arrangement to a more conventional "back-to-back" arrangement that is used to compare displacement sensors and in particular interferometric sensors. In such a setup the two sensors monitor the displacement of a common target from opposite sides. In this arrangement, the frame that connects the references of the two sensors (beamsplitters in the case conventional interferometers) is part of the metrology loop and any changes in the dimensions of this frame are indistinguishable from the measurand. Thus, thermal effects and vibrations in general become part of the measurement and are not common to the measurement made by both sensors. In contrast to the arrangement described above, the arrangement described here excludes the structure joining the two sensor from the metrology loop in such a manner that virtually all of the rigid-body motions of the mechanical support structure, the target mirror and the mount of the SUT become part of the displacement measured by both sensors, allowing for a more faithful comparison. For instance, motion of the SUT mount is measured by both sensors and affects both measurements equally. The metrology loops and the structural loops are largely decoupled, with most of the supporting staging and hardware being outside the metrology loop. Angular motions are also completely rejected in the absence of an Abbè offset. However, since the Abbè offset cannot be entirely eliminated, a compensation scheme based on the measurement of the rigid-body rotations by another interferometer is used to correct for any residual Abbè offset

contributions. This compensation is discussed in detail below.

While this arrangement provides a relatively compact metrology loop, the contribution of every component within the loop must be estimated and included in the estimate of the measurement uncertainty. Thermal changes in the dimensions of the mount, instabilities in the mounting of the reflectors, etc. are in principle within the metrology loop, although in practice the good temperature control over the short duration of the measurement (typically a few seconds) and the use of low expansion materials, result in negligible uncertainty contributions. For example, the effect of target mirror expansion (which is within the metrology loop) is minimized through use of a low expansion material (fused silica) and stringent temperature control (~ 10 mK). The relatively long air-paths on the VM side experience common mode optical path length changes to first order and exhibit adequate performance if shielded from turbulence. A larger contribution results from the unbalanced DPML configuration due to global index changes. A particular example of this effect is apparent displacement due relatively quick pressure changes from human traffic, opening and closing of doors, etc. In practice, the effects of this imbalance can be mitigated to acceptable levels by making relatively quick measurements (over a few seconds).

TABLE 1. Uncertainty budget for the validation metrology

Source	Value (nm)
Environmental effects	0.19
Abbè offset (compensated)	0.12
Wavelength uncertainty	0.03
Slide error motions	0.02
Resolution	0.01
Frequency stability	0.01
Alignment	0.001
Combined std. uncertainty	0.22
Expanded uncertainty (k=2)	0.44

Uncertainty budget

The major sources of uncertainty associated with the measurement of the displacement of the SUT relative to the target mirror by the VM are listed in Table 1. The analysis presented in based on the methodology outlined in the ISO Guide to the Expression of Uncertainty In Measurement (GUM) [2]. Practically all the

sources of uncertainty are represented by uniform distributions and most values of influence quantities in the discussion that follows are typically the half-widths of these distributions. The standard uncertainty is explicitly stated in case of exceptions.

Each of the major contributors is discussed in detail below. The uncertainty quoted for each of the contributors is the standard uncertainty.

Environmental effects

This term includes the effects of environmental variables such as pressure, temperature and humidity. The primary contribution results from the effect of the pressure and temperature on the refractive index. The major contribution is due to global temperature and pressure changes during the course of the measurement. This results in an apparent displacement uncertainty of 0.14 nm due to the relatively large deadpath (20 mm) in the interferometer setup and global pressure and temperature variations of 10 mK and 20 μ m Hg respectively during a typical measurement that lasts a few seconds. The next largest contributions of 0.10 nm results from thermal expansion of the retroreflector mounts, expansion of the target mirror, temperature coefficient of the interferometer and turbulence. Temperature variation of 10 mK is assumed. The remaining contribution of 0.06 nm results from the uncertainty associated with the correction of laser wavelength for the refractive index of air, which in turn arises from the uncertainty associated with the measurement of environmental parameters.

Abbè error compensation

The target mirror is mounted to a linear motor driven crossed-roller linear bearing stage. This stage exhibits two arcseconds and 0.4 arcseconds of pitch (θ_x) and yaw (θ_y) respectively over the scan range. These angular error motions result in an Abbè error due to the offset between the lines of measurement of the SUT and the VM. This offset is in general unknown, but the effects of the pitch and yaw can be compensated for by performing a linear fit to the portion of the measured difference that is correlated to the measured angular error motions. This contribution constitutes a significant source of uncertainty if left uncorrected. After correction, the uncertainty due this contributor is then reduced to the uncertainty in the compensation. Typical Abbè offsets are ~ 0.2 mm and it is estimated that the

uncertainty in the calculation of the offset is 10% or 0.02 mm. In combination with angular error motions of 2 arcseconds, the uncertainty in the compensation contributes 0.12 nm. This includes a contribution due to the uncertainty in the measurement of the angle which results in a contribution of 20 pm.

Wavelength uncertainty

The refractive index based correction of wavelength and the associated uncertainty has been described in the section on environmental effects. This correction is based on the vacuum wavelength of the laser which also has an uncertainty associated with it. The nominal uncertainty ($k=3$) in the wavelength of the stabilized HeNe laser used (Zygo ZMI 7712) is 0.16 ppm resulting in a standard uncertainty of ~ 0.05 ppm. This wavelength uncertainty results in a displacement measurement uncertainty of 0.03 nm. Another facet of the wavelength uncertainty is its stability. The long term (24 hour) wavelength stability of this laser head is 0.3 ppb, which results in a 10 pm contribution.

Slide error motions

The contributions due to the angular error motions (pitch and yaw) of the slide as a result of the Abbé offset have been discussed above. Other contributions that result from the straightness and roll error motions (rotations about the beam axis) are discussed here. Additional contributions due to pitch and yaw are also discussed here.

Straightness error motions couple into the measurement through two effects: Wedge in the target mirror and out-of-flatness of the target mirror in the region where the beam from the SUT impinges on the target mirror. The two straightness error motions are estimated to be one μm over the 0.5 mm travel. In conjunction with a wedge of 5 arcseconds in the mirror, this results in a contribution of 0.02 nm. The contributions of pitch and yaw due to two other factors have also been considered and are found to be negligible. These factors are beam shear and optical path length change due to changes in the angular deviation of the return beam. Although the reflectors in this application are retroreflectors, a residual angular deviation of 0.1 arcseconds is assumed which stems from the deviation from 90 degrees of the retroreflector dihedral angle.

Alignment

The primary alignment term considered here is the alignment of the DPML beam to the direction of motion. Misalignment causes a cosine error which for this travel range of 0.5 mm is negligibly small (~ 1 pm) for a relatively coarse (and easily achievable) alignment tolerance of 10 arcsec.

Periodic errors

Contributions from periodic (cyclical) errors are conspicuous by their absence in Table 1. Periodic errors result from polarization mixing and are typically observed in most polarization encoded interferometer designs. Contributions due to periodic errors do not appear in this uncertainty analysis because of the unique design of the DPML. The measurement and reference beams are spatially separated (unlike many other interferometer designs) and routed in such a manner through the interferometer that virtually eliminates first-order polarization mixing. This behavior was confirmed experimentally by monitoring the output of the Cyclical Error Correction (CEC) functionality of the measurement boards used in the setup (Zygo ZMI 4004), while rotating the incoming polarization states out of alignment by an amount that would have resulted in a significant periodic error contribution. Virtually no periodic errors were observed.

SUMMARY

A description of the validation metrology used to validate the performance of a new high-performance displacement sensor has been described along with the results of a detailed uncertainty analysis which shows that the system is capable of displacement measurement with an expanded uncertainty ($k=2$) of ~ 0.5 nm or ~ 1 ppm.

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SYNTHESIS OF MULTI-AXIS SERIAL FLEXURE SYSTEMS

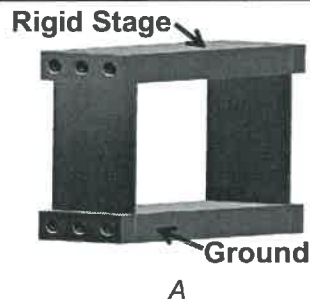
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INSTRUCTIONS

In this paper we show how to design optimal serial precision flexure systems for a given set of specifications —kinematics, range, stiffness, load capacity, and thermal stability. Serial flexure systems consist of multiple flexure elements chained together in series. Figure 1 contrast a serial flexure system with a parallel flexure system.

Parallel Flexure Systems:



Serial Flexure Systems:

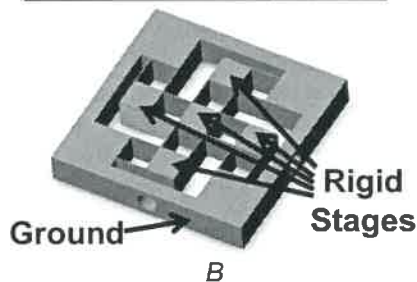


FIGURE 1. (A) Parallel flexure systems consist of a single rigid stage connected directly to ground by flexible constraints. (B) Serial flexure systems consist of multiple rigid stages connected in series and/or parallel by flexible constraints.

Serial flexure systems (i) may possess degrees of freedom (DOFs) that are not possible with parallel systems, (ii) exhibit larger range to size characteristics, and (iii) enable cancelation of parasitic errors. Designers currently rely on motion visualization to generate serial flexure concepts. In many cases, experienced designers have difficulty using these methods to

visualize the motions of serial flexures. This paper provides a synopsis of new flexure design rules and geometric shapes that may be used to more easily visualize and synthesize serial flexure systems.

FUNDAMENTAL PRINCIPLES

We provide a synopsis of key principles that are important to understanding our method.

1st – Hale observed that flexures connected in parallel retain the DOFs that the individual constraints have in common, whereas flexures connected in series will possess the DOFs of the individual constraints combined [1].

2nd – The DOFs of precision flexure systems may be described by geometric shapes that contain screw lines [2-3]. Each screw line represents a single permissible motion for a flexure system. A zero pitch screw represents a rotation about the screw's axis and a screw with large (e.g. infinite) pitch emulates a translation along the screw's axis. If the pitch of the screw is any other value, the motion is a coupled rotation and translation along the screw's axis.

Consider the parallel flexure system shown in Fig. 2. Its DOFs – two rotations of the central stage – may be described by a single geometric shape that contains rotation lines (i.e. screw lines with pitch values of zero). This shape is a disk or pencil of rotation lines that share a common plane and intersect at a common point as shown in Fig. 2A. Figure 2B shows a DOF associated with one of these lines. Geometric shapes that represent the DOFs of a parallel flexure system are uniquely paired to geometric shapes that represent the flexure's constraints.

Consider the geometric shape of constraint lines uniquely paired with the disk of rotation lines shown in Fig. 2C. This shape consists of a sphere and a plane. The sphere represents constraint lines that intersect a common point on a plane. The plane represents any constraint line that also lies on the plane. The significance of this shape is that the constraints of a parallel

flexure system with DOFs described by a disk of rotation lines will lie within this shape. The constraints of the parallel flexure system in Fig. 2 lie within this shape. Figure 2D shows that constraints 2, 3, 6, and 7 lie in the sphere and constraints 1, 4, 5, and 8 lie on the plane.

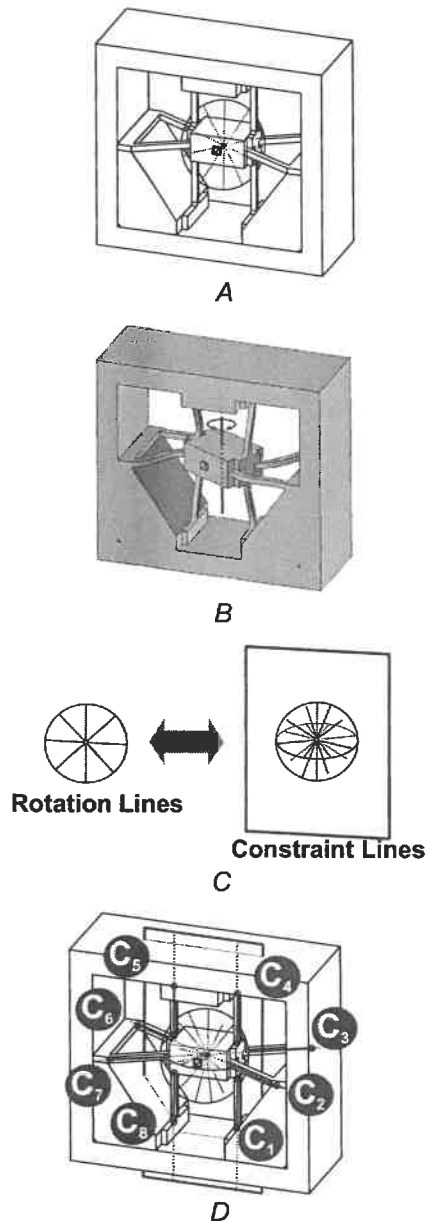


FIGURE 2. The linking of DOFs and constraints via geometric shapes.

Twenty-six pairs of shapes exist that represent possible parallel flexure system layouts and the DOFs they permit [2-3]. As such, there are only 26 'ways' a parallel flexure system may move. To achieve more DOFs, parallel flexure systems

must be stacked in series. The geometric shape that represents a system that possess three and only three translations, for example, may only be achieved by stacking parallel flexure system modules in series. The remainder of this paper discusses very briefly how geometric shapes may be used to generate chains of parallel flexures that yield serial flexure systems with desired characteristics.

DESIGN PROCESS

A parallel and serial version of a three DOF precision flexure will be created and contrasted to illustrate the steps of the process. Our objective is to design a flexure that permits rotations of a probe about its tip. This type of flexure is needed in scanning probe microscopy.

Step 1:

Determine the geometry of the stage and specify the desired DOFs. The geometry of a possible probe holder with the desired rotations is shown in Fig. 3A.

Step 2:

Identify the geometric shape that contains the DOFs from Step 1. The sphere in Fig. 3B is the shape that contains the three rotation lines from Fig. 3A. It represents every rotation line that intersects the tip of the probe. This shape is uniquely paired with a sphere of constraint lines that intersect the same point as shown in Fig. 3C. This pairing defines the relationship between the system DOFs and the constraints that permit them.

Step 3:

Decide whether to create a parallel or serial flexure system. For a parallel flexure system, see Step 4a. If a parallel flexure system is not an option, or if the designer wishes to create a serial flexure system, the designer must continue on to Step 4b.

Step 4a (Parallel flexure systems only):

Select a suitable number of constraints from the geometric shape that contains constraint lines. Hopkins [3] discusses how to select constraints that are independent. For our example, Figure 3D shows three constraints from the sphere that don't lie on the same plane and are therefore independent. A practical embodiment of this flexure system is shown in Fig. 3E.

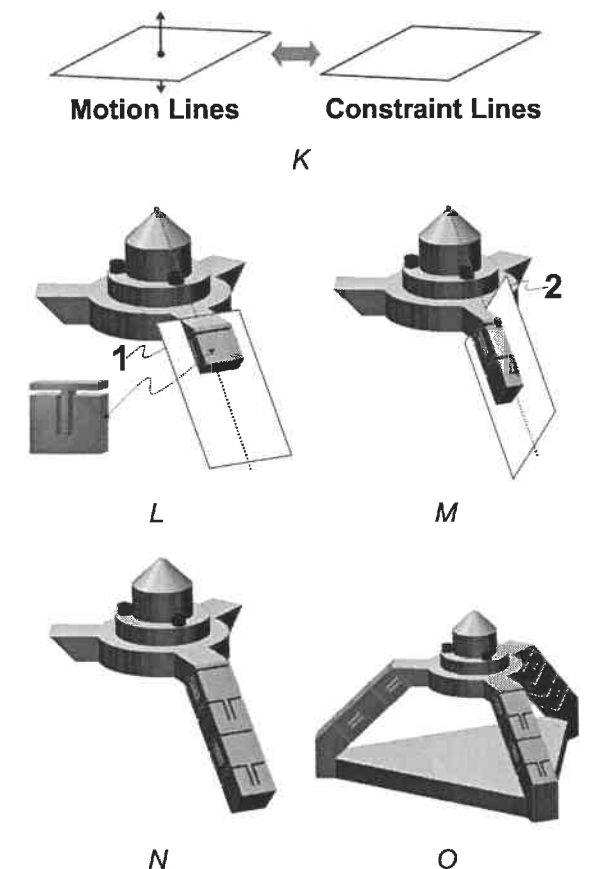
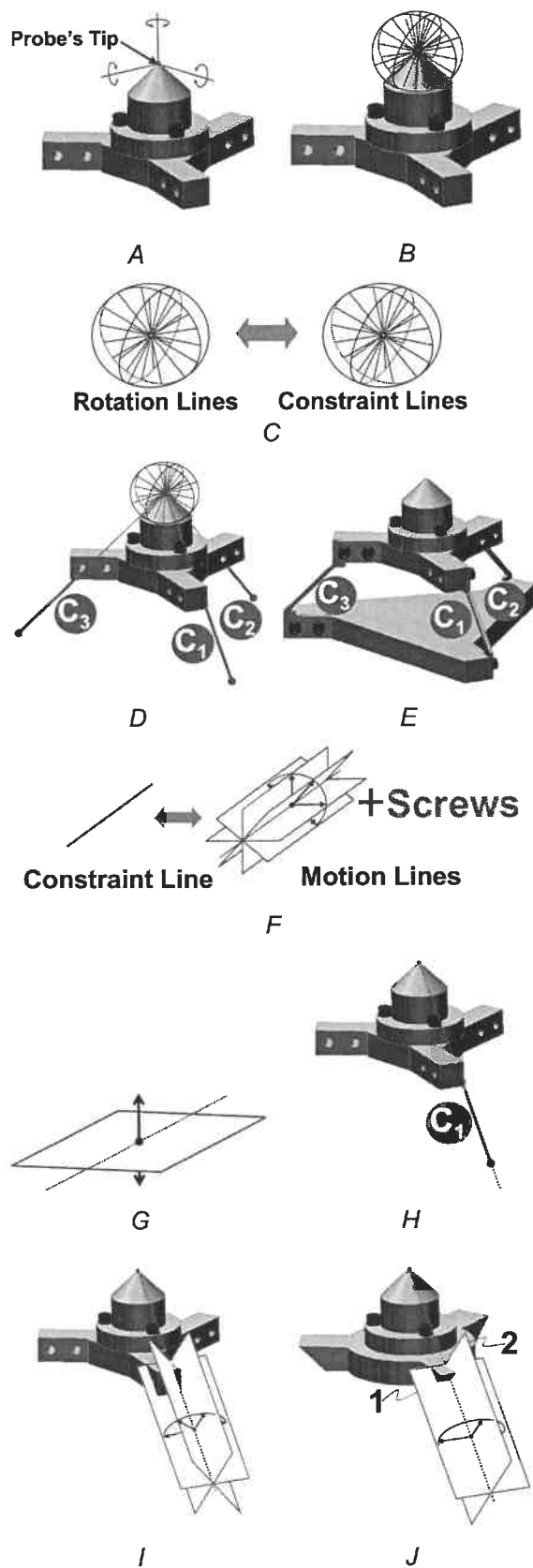


FIGURE 3. Designing a parallel and serial version of a three DOF flexure system.

Step 4b (Serial flexure systems only):

Identify all geometric shapes from the list of 26 pairs that lie within the geometric shape from Step 2. The designer then selects an appropriate location and orientation for each shape relative to the stage. The number, orientation, and location of these shapes affect the range, manufacturability, thermal stability, load capacity and cost of the final flexure design. Next, their uniquely paired constraint shapes are identified and the designer selects constraints from these shapes. The constraints selected from each constraint shape connect stages in the chain of flexure-stage-flexure-stage-etc... This step is somewhat abstract; therefore we will return to this step and apply it to our example shortly.

Step 5 (Optional):

In some cases, a designer may wish to transform the flexure elements within a flexure system into a serial chain of flexure elements in order to enable larger range of motion, increase load capacity, prevent buckling, etc.... The key

is to replace the flexure element with a serial chain of flexures that exhibit the same constraint characteristics.

For our example, we wish to transform constraint C_1 shown in Fig. 3E and 3H into a serial chain of flexure elements. First, we ascertain the shape that describes the DOFs that the replaced element permits. This shape belongs to one of the 26 pairs. For our example, this shape is shown in Fig. 3F. To understand this shape, consider the shape shown in Fig. 3G. This shape is a plane that consists of every rotation line that lies on the plane and a translation perpendicular to the plane represented by an arrow. The shape of Fig. 3F consists of the intersection of these planes along Constraint C_1 's line of action. We replace Constraint C_1 , shown in Fig. 3H, with the shape that describes its DOFs shown in Fig. 3I.

Step 4b should then be performed to create a serial flexure chain. To create this chain, parallel flexure modules must be connected in series such that the combination of their DOFs, lie within the shape shown in Fig. 3I. If, for instance, perpendicular flexure blades were stacked on top of each other like the serial chain shown in Fig. 3N, the combined DOFs from each flexure blade would lie within the shape in Fig. 3I. In order to generate this particular serial chain or other serial chain concepts using step 4b, the reader must understand the 26 pairs well enough to identify shapes that lie within the shape from Fig. 3I. This process is beyond the scope of this paper but will be explained in detail in a later journal paper. For our purposes, we select the shape from Fig. 3G that we know lies within the shape of Fig. 3I. This shape is selected because it is likely to produce flexure blade elements that are easy to manufacture and possess good stiffness characteristics.

We must now select the number of shapes from Fig. 3G to apply and then optimally orient them with respect to the stage. When two of the shapes from Fig. 3G are perpendicularly oriented, as shown in Fig. 3J, and added in series, they yield the DOFs that Fig. 3I represents.

We must now identify the uniquely paired constraint shape of the shapes from Fig. 3J for selecting constraints that will form the serial chain. The shape that contains the constraint lines we need is also a plane of constraint lines

as shown in Fig. 3K. The constraints of a flexure blade would lie on this plane as shown in Fig. 3L. We may, therefore, arrange two flexure blades in series, and perpendicular to each other, as shown in Fig. 3M. The serial chain in this figure possesses the same DOFs as the constraint shown in Fig. 3H. Additional blades may be added in series with the first two, as shown in Fig. 3N, to enable a larger range of motion while maintaining the same constraint characteristics. The other two constraints, C_2 and C_3 , from Fig. 3E could then be replaced by this serially conjugated element as shown in Fig. 3O.

Compared to the parallel flexure version, the serial flexure version of the probe holder possesses a greater load capacity and may achieve a greater range of motion in a smaller space with less susceptibility of buckling. In this instance, therefore, the serial flexure concept from Fig. 3O is better than the parallel flexure concept from Fig. 3E.

CONCLUSIONS

In this paper we have introduced an approach for the early-stage generation of parallel and serial flexure system concepts via the use of geometric shapes. Due to space limitations, a comprehensive description was not possible, and so the concepts were described in the context of a design problem for probe manipulation. Full understanding of the process requires (i) a deep understanding of the principles and practice of Freedom and Constraint Topology in flexure design [2-3] and (ii) practice using the shapes in parallel and serial flexure design.

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NOISE REDUCTION AND DISTURBANCE REJECTION AT THE SUB-NANOMETER LEVEL

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ABSTRACT

This paper examines the noise and disturbance sources that bound the positioning resolution on the Sub-Atomic Measuring Machine (SAMM). These sources are common to many ultra-precision machines.

The SAMM is uniquely designed to position and measure with sub-nanometer resolution over a volume of 25 mm x 25 mm x 0.1 mm. The custom laser interferometer sensors that monitor the x, y, and theta z degrees-of-freedom on the SAMM have been upgraded from a resolution of 0.1 nm to less than 10 pm [1, 2, 3]. By reducing the sensor resolution an order of magnitude, noise and disturbance signals that were previously below the sensor lower bound now dominate the feedback signal and limit the positioning resolution. Since the SAMM is a magnetically levitated stage, positioning in 6 degrees-of-freedom requires a closed-loop controller for stability. The position holding capability represents the positioning resolution and is used to quantify the noise and disturbance influence on the measurement. This paper presents a collection of experiments aimed at identifying the major disturbing agents limiting the position resolution of the SAMM.

INTRODUCTION

As precision instruments continue to evolve and progress toward previously unachievable levels of performance, the knowledge of external disturbances becomes essential. This topic is best summarized by Maxwell, "In designing an Experiment the agents and phenomena to be studied are marked off from all others and regarded as the Field of Investigation. All agents and phenomena not included within this field are called Disturbing Agents, and their effects Disturbances; and the experiment must be so arranged that the effects of these disturbing agents on the phenomena to be investigated shall be as small as possible" [4].

The major disturbing agents applied to this experimental case include:

1. Environmental
 - a. Temperature
 - b. Pressure
2. Acoustic
3. Vibrations
4. Electric and magnetic fields.

These effects may be self-induced meaning that the instrument generates a thermal gradient or excitation force or they may be a result of the environment where the instrument is placed.

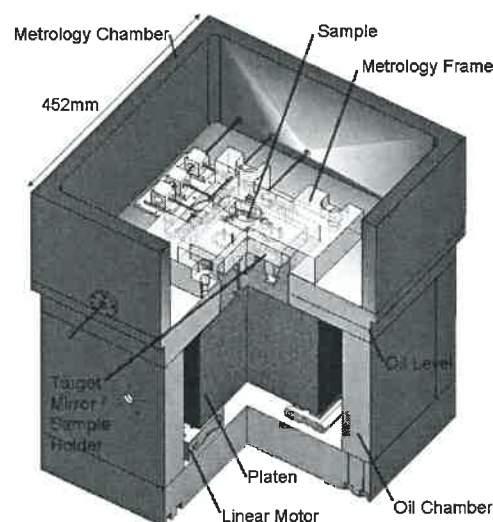


FIGURE 1. Section view of the Sub-Atomic Measuring Machine showing the major components.

BACKGROUND

The Sub-Atomic Measuring Machine is a custom instrument developed by M. Holmes through collaboration between researchers at UNC-Charlotte and MIT [5]. The long range scanning stage has a neutrally buoyant platen floated in silicone oil over four linear motors, refer to figure 1. The motors are a novel Halbach array linear motor design that positions the stage in six degrees-of-freedom [6]. The Zerodur target mirror and measurement sample assembly is elevated above the oil's surface by a flexure-

based Kelvin clamp coupled to the platen. The metrology frame, also of Zerodur, holds three 4-pass laser interferometers and three capacitance gages which read against the target mirror / sample holder. An enclosed metrology chamber above the oil level provides increased stability and environmental isolation from the laboratory. A closed loop control system positions the stage in the travel volume.

The forces for the position control come from the linear motor drives. These drives are more affected by disturbances than other drive options [7]. Robustness against disturbance forces is added in the form of neutral buoyancy of the platform.

LASER INTERFEROMETER SENSOR NOISE

The custom laser interferometers used for position feedback in the control system are primarily responsible for the precision of the instrument. The measured noise of the detection electronics, converted to length equivalent units, illustrates that the interpolation and detection scheme achieves the theoretically calculated resolution of approximately 8 picometers, see figure 2 [2]. The data includes all electronic noise sources in the detection with the exception of the detector.

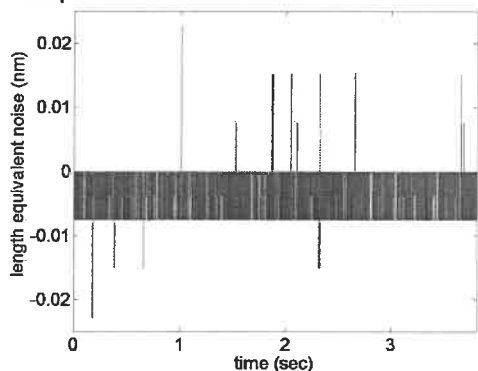


FIGURE 2. Length equivalent noise for the laser interferometer electronics, including all electronics sources except detector.

The detector increases the noise in length equivalent units to 15 pm RMS, see figure 3. The dominant frequency components are 60, 120 and 180 Hz. At this point it is uncertain where this coupling originates. The laboratory lighting and the detector power source have been ruled out as possible sources. The detectors in use are Zygo 7081's, which have been modified to provide a sinusoidal output signal. Perhaps replacing the detectors with more modern options such as photomultiplier

tubes would provide lower noise. Nonetheless, the data presented in figure 3 is considered to be the interferometer sensor noise floor.

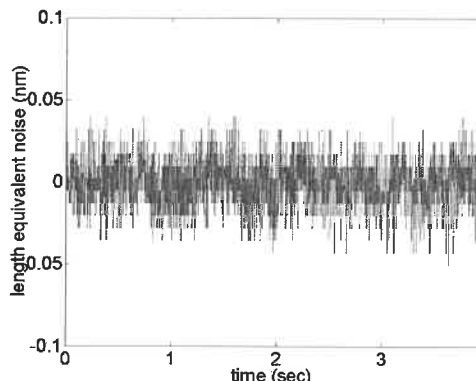


FIGURE 3. Length equivalent noise for the laser interferometers, including detector and electronics sources. RMS = 15 pm.

EXTERNAL DISTURBANCES

The sensor noise limits the performance but disturbances add to and many times dominate the feedback signal. It is desirable to match the position noise to the sensor noise.

Environmental Disturbances

Tight control of environmental disturbances is very important for ultra-precision instruments. Laser interferometers are particularly sensitive to changing environmental conditions in terms of noise and accuracy.

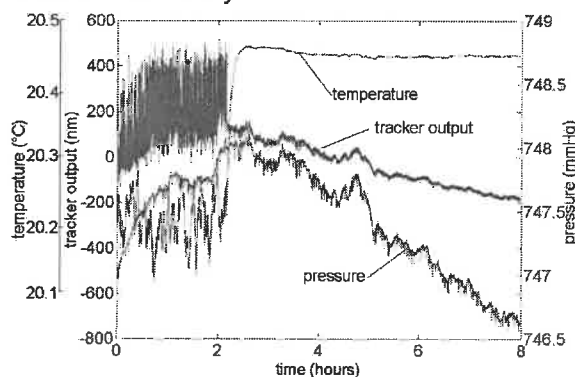


FIGURE 4. Measured temperature, pressure and tracker response due to sealing the metrology chamber.

The metrology chamber on the SAMM is designed to decouple the changing laboratory conditions from the environment where the metrology takes place, see figure 4. The laboratory has temperature control of 20 ± 0.1 °C. At the beginning of this experiment the top cover is removed and a temperature variation of ± 0.1 °C is observed. At this point, this

temperature disturbance dominates the wavelength tracker sensor noise. After 2 hours the chamber is sealed and the temperature inside the metrology chamber rises and a steady state temperature is reached. The inside temperature is elevated by approximately 0.2 °C over ambient. The noise on the wavelength tracking interferometer decreases as the large temperature variation is damped out by the chamber.

The heat load from the linear motor drives is not addressed in this experiment. This data was taken in air with no heat load inside the chamber or the motors operational. The motor commutation scheme minimizes the total power of the motors however experiments have shown that the heat generated by the motors induces a temperature rise in the chamber which will be in addition to the rise due to sealing the chamber [8]. Also these experiments were conducted in air and the instrument will ultimately operate in helium. Helium has numerous attractive qualities including increased thermal conductivity to reduce thermal gradients and nearly an order of magnitude reduction in the absolute refractivity decreasing the sensitivity to environmental disturbances.

Also in regards to environmental disturbances, the offset frequency locked lasers now on the SAMM were so affected by pressure variations that custom housings were necessary for the laser tubes [9]. The slow temperature control of these lasers could not compensate for the fast pressure variations in the laboratory. In order to meet the stability specifications required of these lasers, mechanical decoupling was required. This is an important lesson for precision machines since passive techniques for disturbance rejection can reduce the control effort and hence reduce the heat generated at the actuators.

Acoustic Disturbances

Perhaps the hardest disturbance to address is those from acoustic sources. This is because acoustic energy is broadband in most cases [10]. Mechanical vibration disturbances are more easily found because the frequency of interest can be tracked to a specific source in most cases. Considering acoustic disturbances as broadband sources, most, if not all, vibration modes of a structure can be excited from a single source.

The SAMM is housed in a temperature controlled laboratory. Uniform air flow is distributed from the ceiling and the air returns through hollow walls from the bottom. The fans to distribute the air are audible and generate broadband energy since no significant frequency components can be identified as coming from this source. The effect of the laboratory temperature control on the position performance of the SAMM has been investigated, see figure 5. With the air conditioning (A/C) off, the position noise is 30 pm RMS, twice the sensor noise RMS value. When the A/C fans are restarted the noise increases to 50 pm RMS. This experiment was also conducted in air.

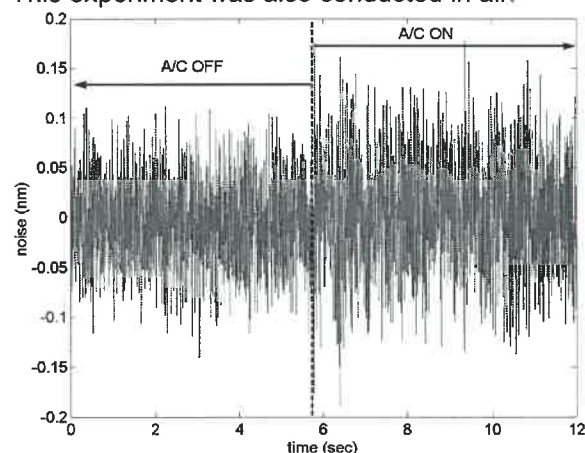


FIGURE 5. Closed loop position noise measurement demonstrating laboratory A/C disturbance. With the A/C off, RMS = 30 pm. With the A/C on, RMS = 50 pm.

Implementing solutions for disturbance rejection can carry consequences. The addition of material to block acoustic disturbances has been considered. The difficulty arises when thermal loads are considered. The temperature rise of figure 4 is with no heat loads inside the chamber volume. Now if acoustic shielding is added around the instrument the heat load of the motors will act to increase the temperatures even more. Acoustic shielding is needed but the execution of this topic remains under study.

Vibrations

Mechanical vibrations also contribute to the position noise. The instrument resides on a commercially available optical table with vibration isolation legs. There is a combination of more than 50 wires, cables, hoses and fibers that couple the table to the floor and the various instruments required to operate the machine. The four six phase linear motors alone account

for 24 2-conductor wire bundles. We suspect this wiring problem accounts for much of the added position noise presented in figure 5. Other sources of position noise include alignment induced noise, environmental induced noise and controller induced noise. Real motions of the reference mirror, the moving mirror or the metrology frame are also possibilities for added position noise. The damped, neutrally buoyant platen largely isolates the moving mirror from mechanical vibrations but the metrology frame is very susceptible to these motions.

This instrument, like most precision instruments, was designed to have mechanical resonant frequencies well above the region of operation. Through various vibration experiments component resonances have been found that will be addressed. For example, the top plate for the chamber has a resonance around 200 Hz that will be pushed closer to 1 kHz in the upcoming round of upgrades. The optical table also demonstrates some interesting modes. Vibrations experiments aimed to quantify the isolation of the table from ground vibrations found table modes easily excited acoustically. Subsequent impact testing found the mode shape of figure 6 at 280 Hz. Skin modes of this type are quite common for these types of optical breadboard tables. Tuned spring, mass, damper systems will be placed near the corners of the table to absorb the energy.

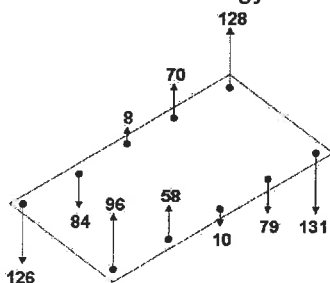


FIGURE 6. Measured optical table mode shape of the 280 Hz resonance, values in nm/N.

CONCLUSIONS

Noise and disturbances are problematic for all precision instruments. The limits of these machines are in direct proportion to the environment. Quantifying the influence on the measurement is a first step in acting to minimize the effect. The dominant disturbances on this instrument remain under investigation and reducing these is essential to enable metrological atomic force microscopy

measurements with sub-nanometer level resolution [11].

ACKNOWLEDGEMENTS

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CARDAN JOINTS WITH AXIS OFFSET IN SINGULARITY FREE HEXAPOD STRUCTURES FOR NANOMETER RESOLUTION

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INTRODUCTION

The main advantages of hexapod systems are user-friendly compact shipment design and high stiffness.

An improvement in the accuracy of hexapod systems can be achieved via better actuator/sensor, cable outlet and joint design.

For most applications the use of external 6 DOF sensor systems at the platform is not suitable due to the sensor performance, the size or the price of such sensor systems. Therefore, sensor systems inside the struts have to be used as a reference for each single axis of the hexapod. The platform position will be determined by the strut-sensor systems, the joints – and the mechanical tolerances (which can be calibrated).

This paper describes different joints used in nanometer repeatable, parallel-kinematic systems. Such joint designs require large computing power of the controller.

HEXAPOD STRUCTURES

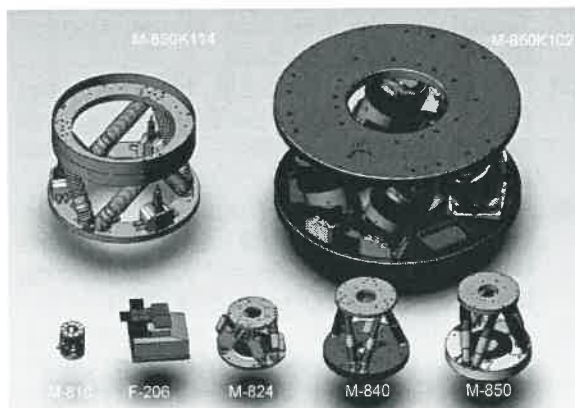


FIGURE 1. Hexapod systems for payload capacity between 10 N and 10.000 N (Physik Instrumente (PI), product examples www.pi.ws)

The multi-axis application defines the structure of the hexapod parallel kinematics. Mostly the customer requirements determine size, height and shaping of the systems. Also environmental

conditions or special scan routines demand different structure design. [1]

Two different parallel-kinematic designs are realized at PI:

Systems with constant length of the struts:

- vertical motion of the lower joints
- horizontal motion of the lower joints

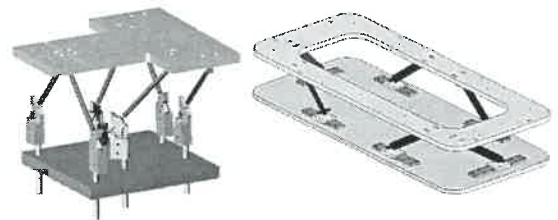


FIGURE 2. a: Vertical moving joints – for small and high load and vacuum flange mounted hexapods. b: Horizontal moving joints – for low height of the position platform

Systems with changeable length of the struts

- Stewart Gough platforms

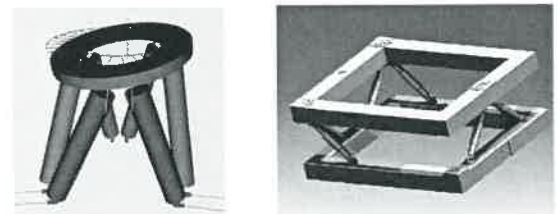


FIGURE 3. Fixed mounted lower joints for small and high load systems with different shaping

The advantage of systems with changeable strut lengths is that no additional linear guiding parts influence the positioning performance. Such systems should be used preferably for high-accuracy applications. However, systems with movable joints could have better dynamic properties because the drives are better decoupled from the platform. All parasitic forces from the drives can be decoupled with the linear guiding.

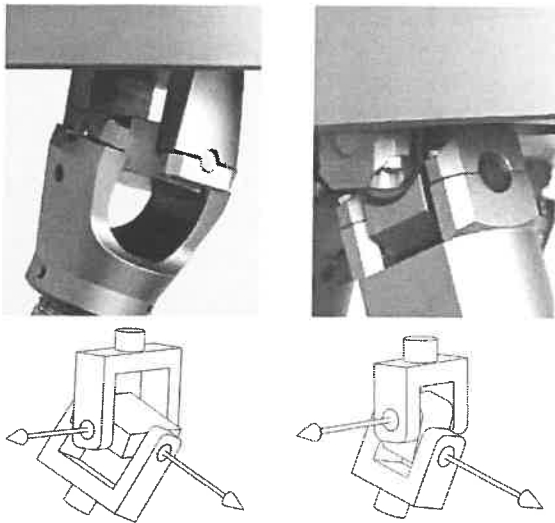


FIGURE 4. a: Example of a cardan joint with axes in one plane, b: Example of a cardan joint with axis offset

DIFFERENT JOINT DESIGNS

The cardan joint with axis offset features a compact joint part between the two links. The two axes can be used as inseparable cylinder parts. In case of high accuracy applications, both bearings inside the joint part (stone) can be realized with long needles. Systems with cardan joints with axis offset provide twice as much stiffness as systems using cardan joint design with crossed axes.

During assembly the cardan joint bearings have to be preloaded or fine adjustment modules should be used for preload. For small angular ranges below 10 degree in each joint, the out-of-plane motion and the roundness of high-precision bearings can reach values better than 50 nm. Hysteresis effects are of high importance. They are caused by the friction and different rolling lines for forward and backward motion. These hysteresis effects can be minimized by medium preload and specially designed spindle bearings.

Sphere joints facilitate the calculation of the inverse kinematics. Our sphere joints consist of ceramics for fully nonmagnetic hexapod systems with piezo actuators. The drawback of such joints is the hysteresis effect due to friction. A very thin lubricant layer is strongly recommended achieving high-precision joints. There are also sphere joints with balls between the joint sockets, but their disadvantage is that boring forces occur during rotation.

Sphere 	Simple mathematics for calculation of the inverse kinematics, friction and hysteresis effects, also boring effects for ball bearing spherical joints
Flexure  	Excellent hysteresis-free designs are possible, problems with non-fixed pivot point, pivot point location depends on the angular motion and load, single part flexures and cardan flexures are integrated
Cardan crossed axis Fig. 4a	High-precision design with preload, medium load conditions
Cardan axis offset Fig. 4b	High-precision design with preload, high-load designs are possible

Flexure design for joints has the advantage of very small hysteresis effects. It can be used for small angular motion of the joints up to 10 degrees. For bigger rotation angles special super-elastic materials are appropriate. The flexure design can also be used for medium load conditions at the joints. The wire flexure is a simple joint which can take in bending as well as torque forces. The drawback of such structures is the unstable pivot point.

INVERSE KINEMATICS OF HEXAPODS WITH CARDAN JOINTS

The hexapod with cardan joints with axis offset can be described as a linkage of six 6R serial linkages systems placed between the base plate and the moving platform. Many different solutions were developed in the 80s and 90s of the last century for the solution of inverse kinematics of a general 6R linkage system. Raghavan and Roth [3] introduced a general algorithm for solving 6R manipulator and related linkages.

$$D = T_1 \cdot G_1 \cdot T_2 \cdot G_2 \cdot T_3 \cdot G_3 \cdot T_4 \cdot G_4 \cdot T_5 \cdot G_5 \cdot T_6 \cdot G_6 \quad (1)$$

Where the transformation matrixes and geometric properties are:

$$T_i = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & \cos(u_i) & -\sin(u_i) & 0 \\ 0 & \sin(u_i) & \cos(u_i) & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix} \quad (2)$$

$$G_i = \begin{bmatrix} 1 & 0 & 0 & 0 \\ a_i & 1 & 0 & 0 \\ 0 & 0 & \cos(\alpha_i) & -\sin(\alpha_i) \\ d_i & 0 & \sin(\alpha_i) & \cos(\alpha_i) \end{bmatrix} \quad (3)$$

A very interesting paper was presented by M. Husty [4] in which he showed that the general problem could be solved by numerical calculation without iteration. The general 6R kinematics of each strut-platform module was cut into two 3R serial chains. The rotation angles α_i of the revolute joints have to be computed.

For the upper part of such 3R-chain systems the direct kinematics can be described as

$$U_1 = T_1 \cdot G_1 \cdot T_2 \cdot G_2 \cdot T_3 \cdot G_{31} \quad (4)$$

$$G_{31} = \begin{bmatrix} 1 & 0 & 0 & 0 \\ \alpha_3/2 & 1 & 0 & 0 \\ 0 & 0 & \cos\left(\frac{\alpha_3}{2}\right) & -\sin\left(\frac{\alpha_3}{2}\right) \\ d_3 & 0 & \sin\left(\frac{\alpha_3}{2}\right) & \cos\left(\frac{\alpha_3}{2}\right) \end{bmatrix} \quad (5)$$

The lower part of the separated 3R-chain can be set to a just equivalent term. A numeric example in the paper shows that the equation can be solved via a simplified chain structure. This method produces sixteen "solutions" for the inverse kinematics.

In our first application in 1993 we used a very similar philosophy but we established an iteration algorithm to solve the two equation parts for the upper and lower linkage system of each strut. Our method provides exactly one solution for each pose. In the past the first algorithm did not rely on the Jacobian matrix for acceleration of the iteration. In the iteration we found a very linear dependence. Also for non pre-settings not more than 3 iteration steps are necessary (typically two steps). We need less than 60 μ s for the direct transformation of all six struts and about 4 ms for the inverse transformation.

It is necessary to indicate that in most of our hexapod systems we use the rotation between spindle and nut for the 5th DOF in each strut. Otherwise an additional precise bearing should be inserted between both cardan joints at platform and base-plate.

Input: The lower plane e collects the spindle axes, the direction vector of joint $\vec{u}$ and the center point of joint P' . In an equal manner plane d collects direction vector $\vec{r}$, point P and the center point of joint Q' .

Output: All joint angles, the distance $P' - Q'$ and the angle between both planes (spindle/nut).

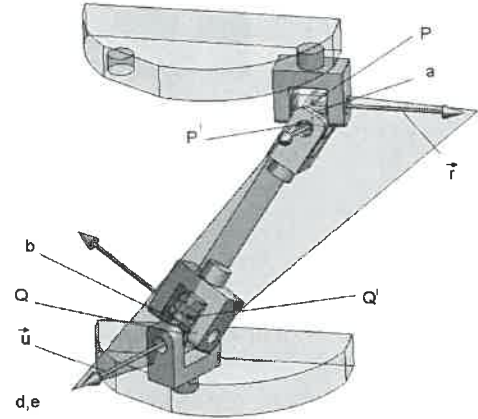


FIGURE 5. Description of variables at general axes configuration

Two equations describe the angular relations of the upper and lower part of the 3R system.

(6), (7)

$$\vec{q} + b(\vec{v} \cos \beta + \vec{w} \sin \beta) = \vec{p} + f\vec{r} + g(\vec{s} \cos \alpha + \vec{t} \sin \alpha)$$

$$\vec{q} + d\vec{u} + e(\vec{v} \cos \beta + \vec{w} \sin \beta) = \vec{p} + a(\vec{s} \cos \alpha + \vec{t} \sin \alpha)$$

The values a and b represent the axis offset. It is assumed that the linkage part axes are rectangular.

With two additional vectors and the scalar product of e. g. $\vec{p} \cdot \vec{w}$ six equations are derived.

Each iteration step calculates the distance between e. g. point P' and plane e . The distance has to be minimized.

$$F_1(\alpha, \beta) = (qps + svb \cos \beta + swb \sin \beta) \sin \alpha - (qpt + tvb \cos \beta + twb \sin \beta) \cos \alpha = 0 \quad (8)$$

$$F_2(\alpha, \beta) = (pqv + sva \cos \alpha + tva \sin \alpha) \sin \beta - (pqw + swa \cos \alpha + twa \sin \alpha) \cos \beta = 0 \quad (9)$$

Equations (8) and (9) lead to two independent constraints for the angles α and β .

$$J(\alpha, \beta) = \begin{pmatrix} \frac{\partial F_1}{\partial \alpha} & \frac{\partial F_1}{\partial \beta} \\ \frac{\partial F_2}{\partial \alpha} & \frac{\partial F_2}{\partial \beta} \end{pmatrix} \quad (10)$$

Jacobian matrix for the two searched angles

$$\begin{pmatrix} \alpha_{n+1} \\ \beta_{n+1} \end{pmatrix} = \begin{pmatrix} \alpha_n \\ \beta_n \end{pmatrix} - J^{-1}(\alpha_n, \beta_n) \begin{pmatrix} F_1(\alpha_n, \beta_n) \\ F_2(\alpha_n, \beta_n) \end{pmatrix} \quad (11)$$

Equation of the iteration algorithm to receive the unknown angles alpha and beta

SOFTWARE SIMULATION TOOL FOR HEXAPOD SYNTHESIS

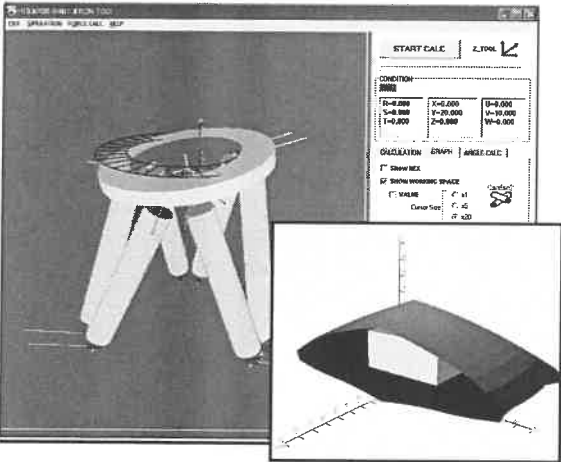


FIGURE 6. Software simulation tool

The simulation program is based on identical routines which also used in our controller. It provides hexapod synthesis, working space analysis, load conditions and joint angle calculation and some scanning features. All types of PI's hexapods with different joint structures are automatically recognized.

MEASUREMENT RESULTS

An optical sphere sensor was developed for the measurement of pivot point stability. A small precise ball is placed at the desired pivot point and fixed at the platform. The 3D sensor housing is placed around this position.

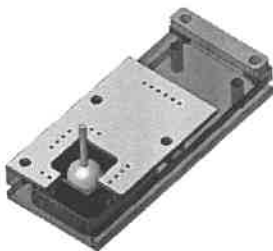


FIGURE 8. Sphere position sensor (3D optical contactless) with resolution of less than 0.1µm

The sensor consists of the optical head and a small industrial computer with three A/D channels. All three deviations from the initial position are reported with command structure. This sensor can also be used for calibration tasks.

The static and dynamic performance of hexapod systems is measured with Zygo laser interferometers in single axes at the platform.

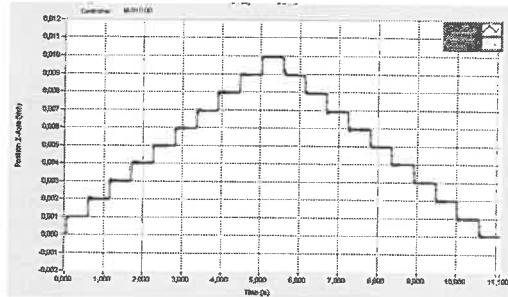


FIGURE 7. Step test protocol for hexapod platform M-810 (repeatability is better than 0.2 µm)

SUMMARY

Major improvements in resolution and repeatability of hexapod systems can be achieved by optimizing the joint design. It is shown that the design of cardan joints with axis offset improve the repeatability of hexapod systems and provide high stiffness. Although cardan joints require much more computing power than simple spherical joints, with standard CPU's the transformation time can be reduced to values less than 1 ms.

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CALCULATING THERMAL LOCATION AND COMPONENT ERRORS ON MACHINE TOOLS

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ABSTRACT

The thermo-elastic deformations of machine tools caused by internal and external heat sources can contribute more than 50% to the overall geometrical inaccuracies of the workpiece [1]. Machine tool manufactures see in mastering thermal effects a key competence on the way to more accurate workpieces. Therefore measurement, numerical simulation and methods for compensation of thermal errors of machine tools moved increasingly into the foreground of latest research activities.

Demonstrative examples illustrate the advantages of observing the entire workspace of machine tools for thermal compensations during operating. Today machine tool controls are able to compute values for compensational movements based on thermally induced component and location errors.

With the effective simulation approach Finite Differences Element Method (FDEM) [2] and the advantages of linear systems of equations it is feasible to compute the spatial compensation values with the introduced method in real time.

INTRODUCTION

Mastering the influences resulting from temperature gradients and temperature changes on machine tools is of capital importance on the way to more exactness on workpieces and to improved process stability during manufacturing. Therefore a lot of work in measurements, numerical simulation and compensation of thermal effects on machine tools is done in latest research activities. Nevertheless all of these attempts are mostly based on just a few positions in the workspace.

Most of the numerical simulations are presented in the past are done with standard finite element software packages. The computation time of transient problems with the finite element method (FEM) is often huge. Kim et al. [3] developed separate models for the machine tool and for the linear scales which turned out to be the main thermal error source.

Wu et al. [4] just modeled the ball screw spindle to compute the thermal effect on a machine centers feed drive system.

In compensating thermal effects two different essential ways are applied. On one hand thermal deformations at the tool centre point (TCP) are reduced via design modifications. In this case commonly no controlled actuators are operating. On the other hand the TCP position is readjusted by controlled actuators [5]. Controlled actuators are commonly the machine tools feed drive system but also other actuators are used. E.g. plane heaters and cooling jackets [6].

The effective simulation approach FDEM allows computing the thermally induced TCP-displacements on machine tools with a three-dimensional physical model of the whole machine tool in real time [5]. Knowing the thermally induced TCP-displacements in the workspace modern machine tool controls are then able to readjust the position with the machine tools feed drive system.

MOTIVATION

It is often reported that compensation models for thermally induced TCP-displacements only consider measurements at one position in the workspace of machine tools. Nevertheless measurements and numerical simulation results of the thermally induced TCP-displacements at different positions in the work space have pointed out that the absolute value and the sign may vary substantially. In Figure 1 the measurement setup used for a five-axis-milling machine prototype to detect the TCP-displacements in several positions of the A-axis and the corresponding measured relative displacements in X-direction are diagrammed.

The second example in Figure 2 shows the positioning error EYY of a three-axis-milling machine at different times during an operation loop. The computations are based on a FDEM simulation model of the machine tool.

Both examples illustrate clearly that it is necessary to consider more than one axis

position if a model is used to compensate thermal effects in the whole machine tool workspace with high accuracy.

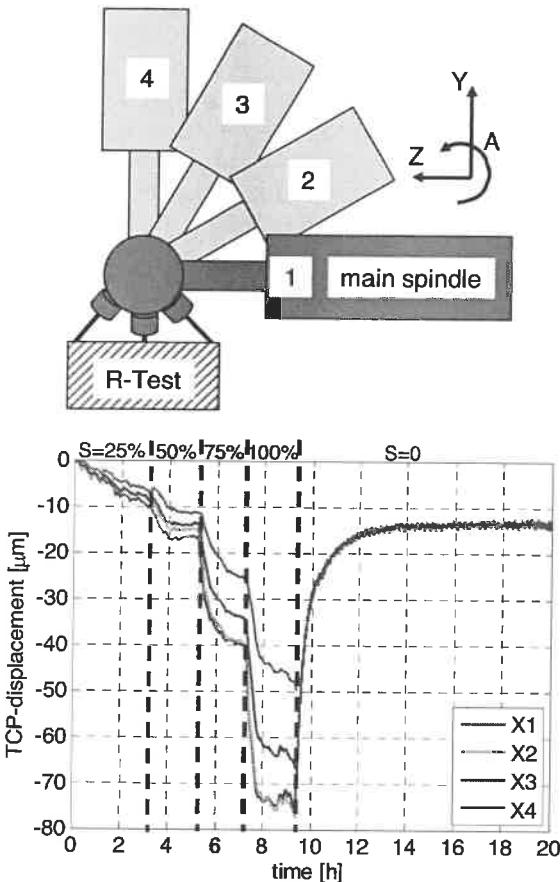


FIGURE 1. Top: measurement setup for a five-axis-milling machine. Bottom: thermally induced TCP-displacements in X-direction caused by rotating main spindle at different speeds (S).

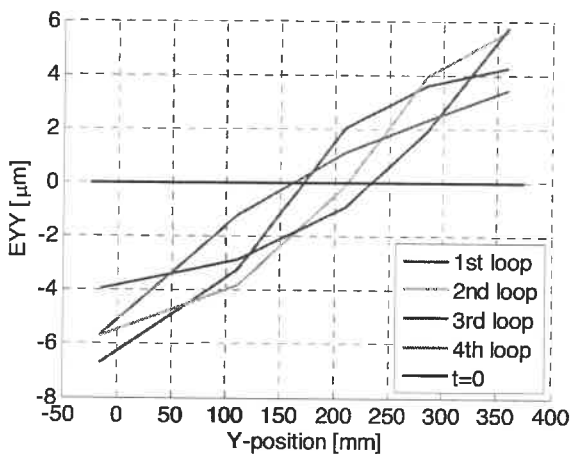


FIGURE 2. Numerical simulation result: Thermally induced positioning error EYY of a three-axis-milling machine without a linear scale during an operation loop.

REDUCING SYSTEMS OF EQUATIONS

For the computation of the thermally induced deformations with FDEM a linear system of equations (1) has to be solved [2,5].

$$\underline{\underline{M}} \cdot \underline{u} = \underline{T} \quad (1)$$

Where $\underline{\underline{M}}$ is the system matrix, $\underline{u}$ are the nodal displacements and $\underline{T}$ is the temperature vector.

To compute the thermally induced deformations usually the whole mechanical structure, in this case the machine tool, is analyzed. In fact, for compensation just the displacements at the TCP are required.

Methods like substructure technique or elastic foundation, known from the Finite Element Method (FEM), allow reducing a large system of equations into smaller systems when just a reduced number of degrees of freedom have to be obtained. Nevertheless using these methods for equation (1) the right hand side vector $\underline{T}$ has to be manipulated before the reduced system can be solved. An alternative way is to create a system like equation (2) where the right hand side vector $\underline{T}$ does not require manipulation before the system can be solved.

$$\underline{u} = \underline{\underline{M}}^{-1} \cdot \underline{T} \quad (2)$$

For large systems the inverse of the matrix $\underline{\underline{M}}^{-1}$ can not be handled with moderate effort. For small systems and if from a large system just a few degrees of freedom are required $\underline{u}$ can be computed very effectively via a matrix-vector multiplication. Considering the second case just the required lines $\underline{\underline{M}}_{red}^{-1}$ of the inverse $\underline{\underline{M}}^{-1}$ have to be computed. $\underline{\underline{M}}_{red}^{-1}$ can efficiently be computed by solving equation (3).

$$\underline{\underline{M}} \cdot \underline{\underline{M}}_{red}^{-1} = \underline{\underline{E}}_{red} \quad (3)$$

$\underline{\underline{E}}_{red}$ are the required columns of the identity matrix.

COMPUTING TCP-DISPLACEMENTS

For a universal milling machine the manufactured workpieces and the used tools are wide ranged and mostly unknown to the compensation model builder. Instead of considering the geometry of the workpiece and the tool a global geometrical model can be used to compute the displacements at the TCP. In Figure 3 the calculation scheme for the tool-side TCP-displacements is illustrated. An equal model is used to compute the workpiece-side

TCP-displacements with known deformations of three nodes of the table.

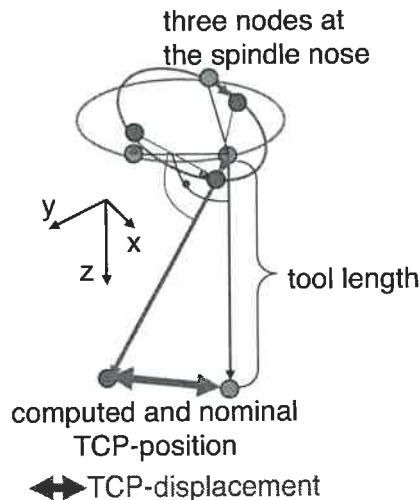


FIGURE 3. Computation scheme of the tool-side TCP displacement based on tool length and the displacements of three nodes at the spindle nose

EXAMPLE 1: THERMAL LOCATION AND COMPONENT ERRORS OF A THREE-AXIS-MILLING MACHINE

Bringmann [7] developed a method to compute the location and component errors of a three-axis-milling machine by measuring the deviations with a 3D-ball plate. His measuring results are the deviations at a 3-D space lattice. Using this method to compute the thermal component and location errors of a machine tool it is necessary to get the thermally induced TCP-displacements in the workspace from different FEM models at a 3D space lattice also. With the FDEM the machine tool is meshed in a way that different positions along the traverse path of the axes can be modeled and the interaction between the elements of the feed drive system parts do not lose there validity. This enables to compute the TCP-displacements at defined axes positions in the workspace with different FEM-models of the machine tool and a given temperature field. The FEM-models are arranged in a way that the computed results give the TCP-displacements in a 3-D space lattice of the machine tool workspace. From the displacements in the space lattice the component and location errors of all three axes are derived.

As illustrative example for the method the positioning error EYY along the Y-axis is explained. The 3-D space lattice is shown on the right hand side in FIGURE 5; caused by the

different axis length five positions are used in X- and Y-, three positions are used in Z-direction. In a first step the Y-deviations along the Y-axis at the corresponding 15 lines in the X-Z-plane are computed. Thermal deformations can change the zero position Y0Y what is representing a Y-movement of the 3-D space lattice according to the machine tool coordinate system. Therefore first the computed EYY of all 15 lines have to be reduced by the average value of the corresponding line.

$$EYY_{new}(X, Y, Z) = EYY(X, Y, Z) - \frac{1}{5} \sum_{i=1}^5 EYY(X, Y_i, Z) \quad (4)$$

The positioning correction value EYY_{cor} of a Y-position is the average of the 15 errors in the X-Z-plane computed with (4).

$$EYY_{cor}(Y) = \frac{1}{15} \cdot \sum_{i=1}^5 \sum_{j=1}^3 EYY_{new}(X_i, Y, Z_j) \quad (5)$$

EXAMPLE 2: THERMAL LOCATION AND COMPONENT ERRORS OF A ROTARY AXIS OF A MACHINE TOOL

A similar procedure described for linear axes is used to compute thermal location and component error for rotary axes. Several FEM-models with different axis positions of the rotary axis are used to compute TCP-displacements in defined positions. The example in Figure 5 illustrates the procedure with the pictured model of a B-axis of a machine tool in Figure 4.

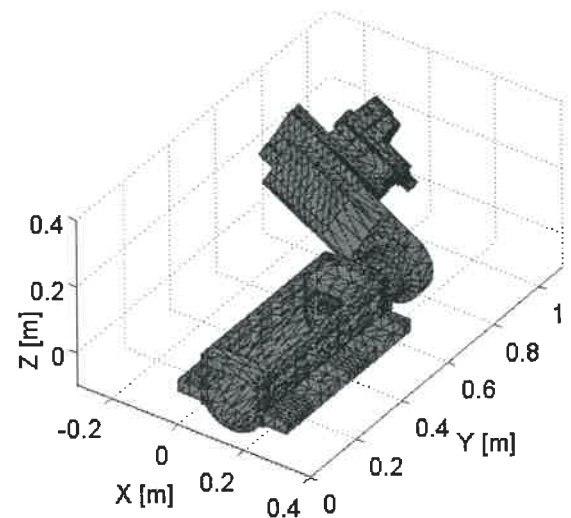


FIGURE 4. Model of the B-axis.

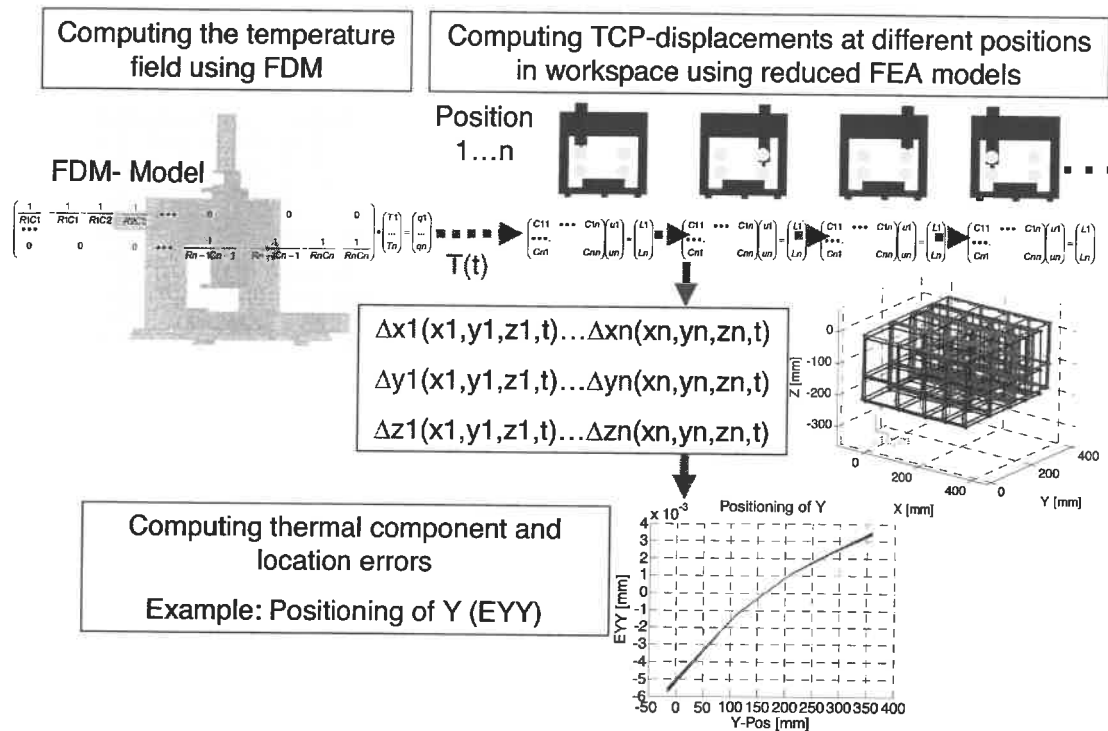


FIGURE 5. Schematic of the procedure: Illustrated by computing thermal component and location errors of a three-axis-milling machine

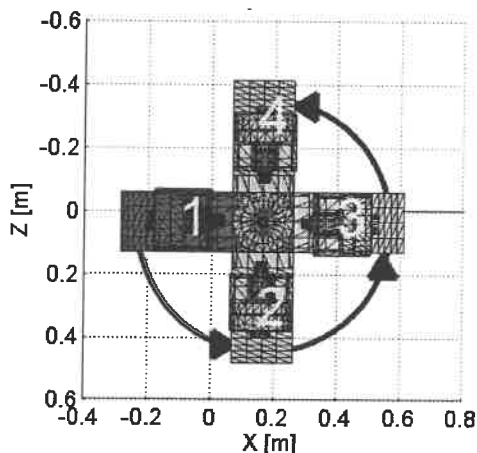


FIGURE 5. Test cycle of rotary axis

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MECHANICAL DESIGN AND PRACTICE OF ARTIFICIAL CHANNEL-CUT CRYSTAL STAGES FOR THE BONSE-HART USAXS INSTRUMENT WITH 10-NANORADIAN-SCALE RESOLUTION AND STABILITY

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INTRODUCTION

Small-angle X-ray scattering (SAXS) is a nondestructive measurement technique in which the elastic scattering of X-rays from inhomogeneities within a sample is recorded at low scattering angles. The typical range of scattering angles in a pinhole SAXS camera delivers structural information on lengths between 1 nm and 100 nm. To resolve microstructures with larger dimensions, however, the smaller angles available with ultra-small angle X-ray scattering (USAXS) are required.

The Advanced Photon Source (APS) USAXS instrument is located at X-ray Operations and Research (XOR) Sector 32, where it receives photons from an APS undulator x-ray source. The undulator source provides continuous access to X-rays in the energy range from 3.2 to above 80 keV. The photons subsequently pass through a fixed-offset Si (111) monochromator capable of tuning the photon energy between 8 and 35 keV. To reject the unwanted higher-order harmonics, a pair of vertically reflecting flat silicon mirrors follow the monochromator. The mirrors have three stripes, Si, Cr, and Rh, each with a different angular dependence on X-ray energy for total reflection. By means of a simple horizontal translation to select the appropriate reflection stripe, the mirrors reject harmonics for primary photons in the energy range from 8 to 19 keV, which is the operational range of the APS USAXS [1,2].

As shown in Figure 1, the APS USAXS instrument in one-dimensional collimated imaging configuration includes 2D slits, a Si (111 or 220) collimating system, an ion chamber, a sample holder stage, a Si (111 or 220) analyzer system, and a photodiode detector system.

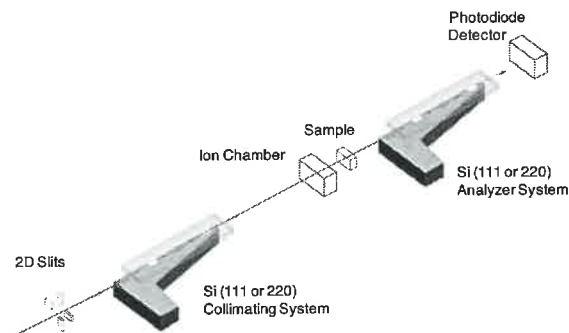


FIGURE 1. Schematic of the APS USAXS instrument in one-dimensional collimated USAXS imaging configuration.

The collimating crystal system and analyzer crystal system are similar, and make use of a design illustrated in Figure 2, where the triangular cut of the first crystal makes it possible to select the number of reflections by translating the crystal pair horizontally with fixed-gap artificial channel-cut crystal geometry.

A windowless ionization chamber after the collimating crystals monitors the X-ray intensity reaching the sample position. The scattering sample is mounted on a standalone stage with two motorized orthogonal translational axes. This allows samples to be attached to a sample-mounting plate and measured in sequential order. Space around the sample stage makes possible the installation of sample environments such as heating stages, flow cells, tensile stages and liquid cells [1].

In this paper we present the mechanical design of the artificial channel-cut crystal stages for the APS USAXS instrument. Finite-element analyses for the laminar weak-link mechanism as well as X-ray test results for the USAXS crystal X-ray optics are also presented.

THE CRYSTAL OPTICS FOR A FIXED-GAP BONSE-HART USAXS INSTRUMENT

Unlike the original and widely adopted design of Bonse and Hart USAXS instruments [3] that used an odd number of reflections from the collimating crystals, the APS USAXS instrument use an even number of reflections in a nondispersive geometry. A narrow crystal gap, combined with the many reflections, prohibits the direct (uncollimated) beam from reaching the sample.

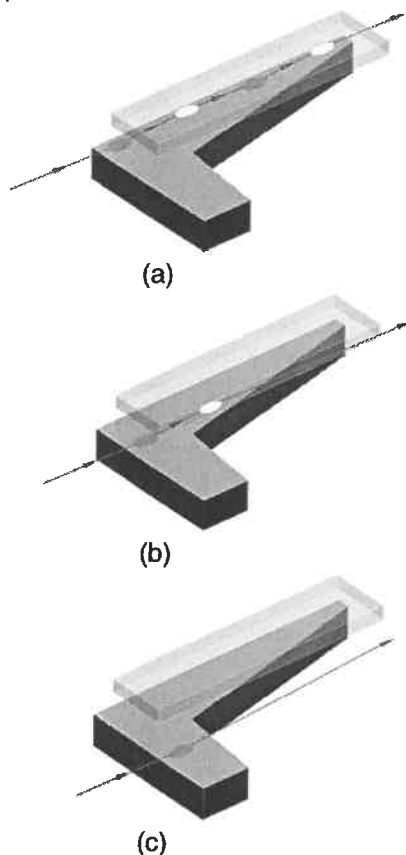


FIGURE 2. Schematic of the APS USAXS fixed-gap collimating crystals. (a) configuration for four reflections. (b) configuration for two reflections. (c) single reflection configuration for commissioning alignment.

THE ARTIFICIAL CHANNEL-CUT CRYSTAL STAGES

The quality of the crystal surfaces is the key to ensuring that the wings of the instrumental rocking curve are effectively suppressed over the large q range for USAXS data acquisition. To overcome the manufacturing difficulties for a

regular channel-cut crystal with small gap and exceptional surface quality, artificial channel-cut crystal stages [4,5] are applied for both collimating crystals and analyzer crystals. The precision and stability of this mechanism allowed us to align or adjust a pair of high-surface-quality crystals to achieve the same performance as that of a single channel-cut crystal, and to ensure the 10-nanoradian-scale alignment resolution and stability required for their variable multiple-reflection X-ray optics in an angular alignment range of near 1 degree.

Design of the artificial channel-cut crystal stage

Figure 3 shows a 3D model of the artificial channel-cut crystal stage for the APS USAXS instrument vertically diffracting crystal analyzer system. As shown in this figure, the base plate (2) of the artificial channel-cut crystal stage is mounted on a commercial rotary stage (11) from Aerotech, Inc. [6] via a pair of adapter plates (12,13). The adapter plates provide manual linear alignment capability between the rotary stage and the artificial channel-cut crystal stage.

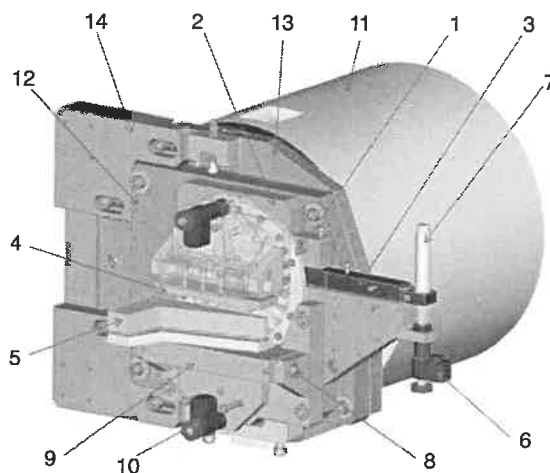


FIGURE 3. A 3D model of the artificial channel-cut crystal stage for the APS USAXS instrument crystal analyzer system: (1) high-stiffness, weak-link mechanism module; (2) base plate; (3) sine-bar; (4,5) two silicon single crystals; (6) Picomotor™; (7) Physik Instrumente™ closed-loop controlled PZT; (8) commercial flexure bearing; (9) crystal holder; (10) Picomotor™-driven structure; (11) Aerotech™ rotary stage; (12,13) adapter plates; (14) balancing plate.

There is a pair of stacked thin-metal weak-link modules (1) used in the driving mechanism. The weak-link module is a planar-shaped, high-

stiffness, high-stability mechanism acting as a planar rotary shaft. One set of weak-link mechanisms is mounted on each side of the base plate (2). A sine-bar (3) is installed on the center of the planar rotary shaft for the pitch alignment between the two single crystals (4, 5). Two linear drivers are mounted on the base plate serially to drive the sine-bar. The rough adjustment is performed by a Picomotor™ (6) with a 20- to 30-nm step size. A Physik Instrumente™ closed-loop controlled PZT (7) with strain sensor provides 1-nm resolution for the pitch fine alignment. A pair of commercial flexure bearings (8) mounted on each of the crystal holders (9) and Picomotor™-driven structures (10) to provide the roll alignment for each crystal. To keep the load balanced for the rotary stage (11), a balancing plate (14) is applied on the adapter plate (13). Figures 4 and 5 illustrate the side, top, and rear views of the artificial channel-cut crystal stages.

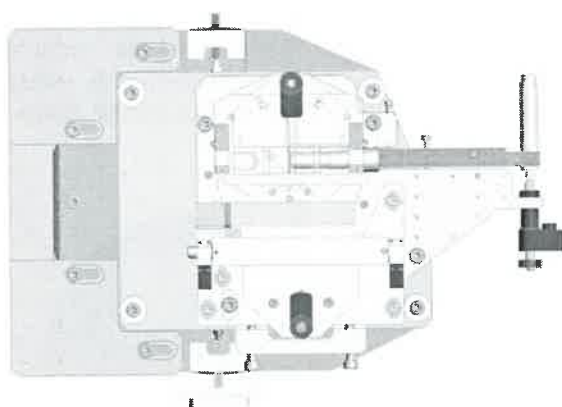


FIGURE 4. A side view of the 3D model of the artificial channel-cut crystal stage for the APS USAXS instrument crystal analyzer system.

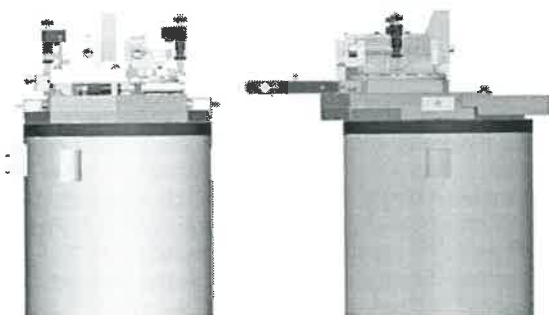


FIGURE 5. The rear view (left) and top view (right) of the 3D model of the artificial channel-cut crystal stage for the APS USAXS instrument crystal analyzer system.

The laminar overconstrained weak-link module

The original laminar overconstrained weak-link module was created to overcome the obstacles encountered in developing a 4-crystal in-line high-resolution hard x-ray monochromator using a nested channel-cut crystal geometry with meV bandpass. The first high-stiffness weak-link mechanism with stacked thin-metal sheets developed for the APS high-energy-resolution beamline 3-ID [4,5]. The precision and stability of this mechanism allowed us to align or adjust an assembly of crystals to achieve the same performance as that of a single channel-cut crystal, so we called it an “artificial channel-cut crystal.”

Unlike traditional kinematic linear spring flexure mechanisms, the overconstrained rotary weak-link mechanism provides much higher structure stiffness and stability. Using a laminar structure configured and manufactured by chemical etching and lithography techniques, we are able to design and build a planar-shape, high-stiffness, high-precision weak-link module as shown in Figure 6. The precision of modern photochemical machining processes using lithography techniques makes it possible to construct a strain-limited overconstrained structure from thin metal sheets. By stacking these thin-metal weak-link sheets with alignment fixtures, we can construct a solid, complex, thick weak-link module with reasonable cost [5].

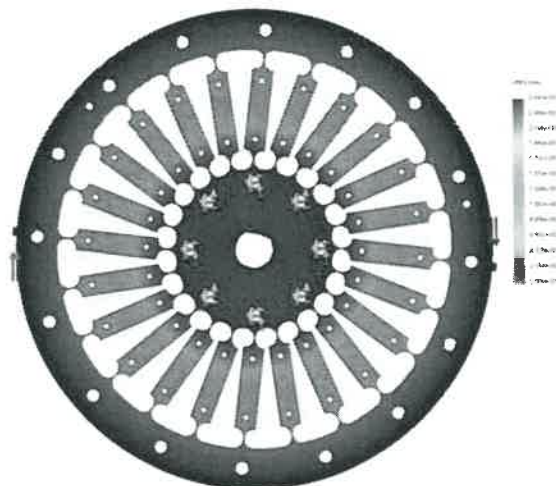


FIGURE 6. A finite-element simulation for a wheel-shaped rotary weak-link module. It shows the displacement distribution under a torsion load on the outer ring while the center part is fixed on the base.

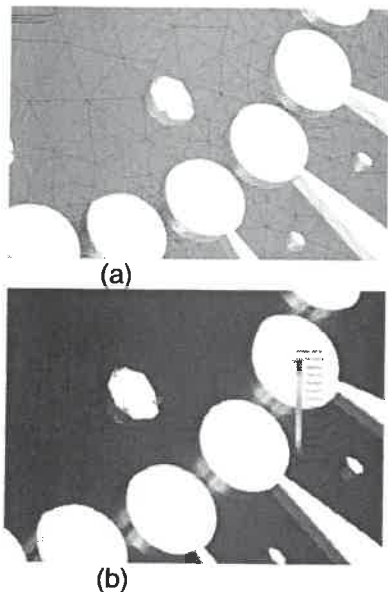


FIGURE 7. Finite-element simulation for a wheel-shaped rotary weak-link module. (a) Detailed mesh for finite element analysis; (b) Stress distribution under a torsion load on the outer ring while the center part is fixed on the base.

As shown in Figures 6 and 7, the overconstrained weak-link module made from 304 stainless steel is capable of performing a ± 0.25 degree rotary elastic displacement with 20% safety margin to the material yield strength. With high-strength materials, such as stainless steel 17-7 ph, the same module will be capable of performing a ± 1 degree rotary elastic displacement with 30% safety margin to its yield strength.

X-RAY TEST RESULTS AND DISCUSSION

The APS Bonse-Hart USAXS instrument using artificial channel-cut crystal stages at the APS XOR Sector 32 at Argonne National Laboratory is now operational. It is a fundamental characterization facility that is optimized for the high brilliance and low emittance of an APS undulator source. Its 10^{-4} angular and energy resolutions, accurate and repeatable X-ray energy tunability over an operational energy range from 8 to 18 keV, and excellent signal-to-noise ratio support a wide range of tasks, from addressing the fundamental scientific issues in soft matter to characterization of shocked materials.

Figure 8 shows the APS Bonse-Hart USAXS instrument crystal analyzer using artificial channel-cut crystal stages at APS XOR Sector

32. Figure 9 shows an X-ray rocking curve scan using the artificial channel-cut crystal stage with a step size of about 300 nrad.

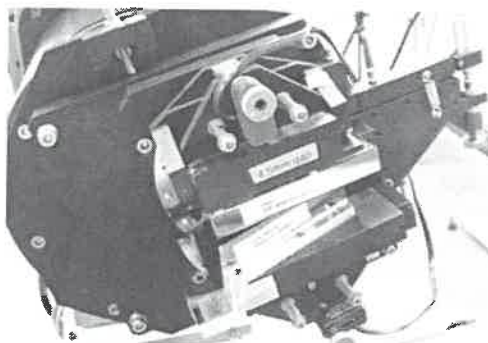


FIGURE 8. Photograph of the APS Bonse-Hart USAXS instrument crystal analyzer.

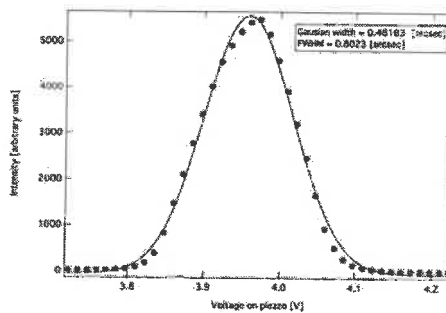


FIGURE 9. An X-ray rocking curve scan result using the artificial channel-cut crystal stage.

ACKNOWLEDGEMENTS

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OPTICAL AND FIDUCIAL FABRICATION WITH A FAST LONG RANGE ACTUATOR (FLORA)

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INTRODUCTION

Diamond turning (DT) has revolutionized the fabrication of optical surfaces for consumer, defense and science applications such as contact lenses, forward-looking infrared radar and infrared spectrometers. It has made this impact not only because it can accurately and rapidly fabricate diffractive, refractive and reflective optical surfaces, but also because it can create reference features tied to the optical surfaces to assist in the assembly process. An emerging trend in optical design is the use of Non-Rotational Symmetric (NRS) surfaces to reduce complexity, bulk and weight while improving optical performance. To create these so-called freeform surfaces, DT machines with linear motors can operate at slow spindle speeds (<20 rpm) or a Fast Tool Servo (FTS) can be added. Early examples of the latter were driven by piezoelectric stacks and had limited stroke (<100 μm). A Fast Long Range Actuator (FLORA) [1-2] has been developed to create NRS optical surfaces with millimeters of sag at machining speeds up to 1200 rpm. Compared to a stationary tool post, the FLORA adds significant capability but, at the same time, can add unwanted motions to the tool that could reduce the surface quality. To minimize these motions, a control system has been developed and demonstrated by machining flat and tilted flat surfaces using different workpiece materials.

OPTICAL AND FIDUCIAL FABRICATION

The ability to machine large-amplitude (> 1 mm) NRS features also makes it possible to create assembly features in the same operation as the optical surface. This is critical for freeform surfaces because the added degrees of freedom complicate assembly.

Biconic Mirror

The M4 mirror was designed by NASA as a crucial component of its IRMOS space-based spectrometer. This mirror shape is used as a demonstration surface in this paper to illustrate the performance of the FLORA system. The mirror is an off-axis biconic toroidal surface that

blends two oblate ellipses with different curvature and eccentricity in the XZ and YZ planes. As such it has no axis of rotational symmetry.

Figure 1 shows a section of the biconic surface as a wire frame and the aperture of the mirror blank as a solid. This optic was originally machined at the PEC [3] in 2002 using a piezoelectric actuator with mechanical amplification for a 400 μm range. Because this range was less than the sag of the M4 surface, the optical blank was machined off-axis and tilted 35.3° with respect to the fabricating axis. The actuator was also tilted with respect to the spindle axis. These two changes allowed the surface to be machined with the range available.

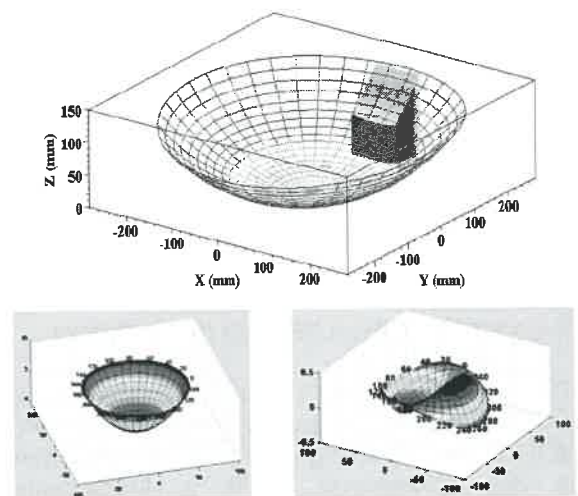


FIGURE 1. Biconic optic M4 showing (top) parent optic, (Low-left) on-axis M4 surface and (low-right) NRS component.

With the extended range of the FLORA, it is feasible to machine the M4 surface on axis. The lower images in FIGURE 1 show the on-axis M4 surface with aperture size of 98 mm with the surface translated -227.41 mm in Y direction and rotated -35.3° around X axis. The total excursion of the on-axis surface is 4.054 mm, but after removing the best sphere with a diameter of

304.881 mm, the excursion of NRS component is 0.545 mm.

Fiducial Features

An example of an ideal assembly feature is the kinematic coupling shown in Figure 2. The right hand part is a blank for an optic with three raised cosine-like features. The left hand part represents the optical housing and has 3 grooves that receive the mirror surface and locate it in 6 degrees of freedom without over-constraint and potential distortion. This concept could replace the time consuming shim-and-measure technique that has plagued the application of freeform surfaces in multi-element optical systems.

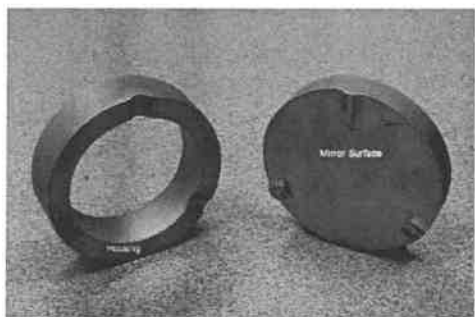


FIGURE 2. Kinematic coupling used to mount freeform optical surfaces in 6 degrees of freedom.

The shape of the mating features is a compromise between the needs for fabrication and operation of the interface. There are two main concerns.

- The stress at the contact should not exceed the yield stress of the material. Since the materials are 6061-T651 aluminum, large radii at the contact point are important.
- The slope of contact surfaces should be sufficient to overcome friction at the interface. There are assumed to be 2 contact points on each of the 3 features, all with identical loading. Therefore, it is important that the protruding features be steep enough to fall to the bottom of the grooves.

Figure 3 shows detailed shape of the feature design. They have constant 4 mm width in the radial direction and a full cosine wave in the circumferential direction with a wave length of 8 mm. The high-point in the radial direction is at a radius of 61.5 mm. The sinusoidal wave has the maximum amplitude of 0.6 mm, but the

amplitude (A) changes as a parabolic function of radial position (r)

$$A = 0.6 - \frac{(r - 61.5)^2}{100} \quad (1)$$

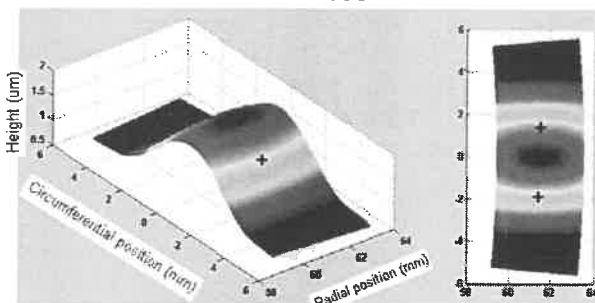


FIGURE 3. Details of the fiducial feature. The theoretical contact points with the v-groove is indicated by the crosses.

The two points of contact are shown in Figure 3 at a radial position 61.5 mm and a phase of 60° from the peak along the cosine shape. The contacts are 2.7 mm apart and 0.3 mm below the top of the feature. The slope at the contact point is 22°, but the maximum slope is 25.22°. The slope was selected to be larger than the critical angle for a pair of mating aluminum surfaces. This was measured to be 9° or a friction coefficient of 0.16. A customized diamond tool with a 27° clearance angle was supplied by Chardon Tool to avoid hitting the back of the tool on the down slope of the cosine wave. This tool was used for fabricating both the optical and fiducial surfaces.

Toolpath Generation

The generation of the toolpath for the optical and the fiducial surfaces are complicated by their NRS shape, the need for feedforward acceleration command to minimize the following error [2] and compensation for tool nose radius.

Axes Selection

A 2-axis diamond turning machine was used for fabrication with the part mounted on a spindle on the Z-axis and FLORA mounted on the X-axis (radial direction on the part). As a result, z motion can come from the Z-axis, the FLORA or both. Because the range of motion of FLORA is sufficient to do the entire surface, the Z-axis was not used in the fabrication.

Tool Nose Radius Correction

The shape of the optical surface and fiducial feature can be represented by a point cloud after

correction for the tool nose radius. This correction, which must be calculated over the entire NRS surface, is illustrated in Figure 4 for the case of a tilted flat. As the part rotates, the normal to the surface changes and with it the contact point of the tool and workpiece. For any part radius (R_p) and θ , the contact angle can be calculated and the corrections δx and δz in Figure 4 can be found. A new set of commands (x, z) will result.

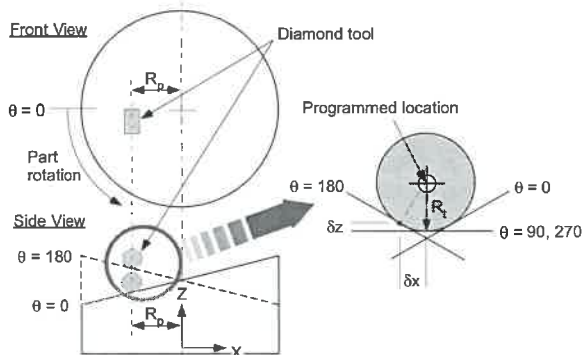


FIGURE 4. Tool Nose Radius correction

Optical Surface

After finding the new tool path with tool radius (0.9977 mm) compensation for the M4 surface in Figure 1, a uniform polar mesh grid of 50x361 (radius, mm x spiral dist., mm) is created and a bilinear interpolation algorithm [3] is applied in real-time to generate position and acceleration commands as $f(r, \theta)$ at the update rate of 20 KHz.

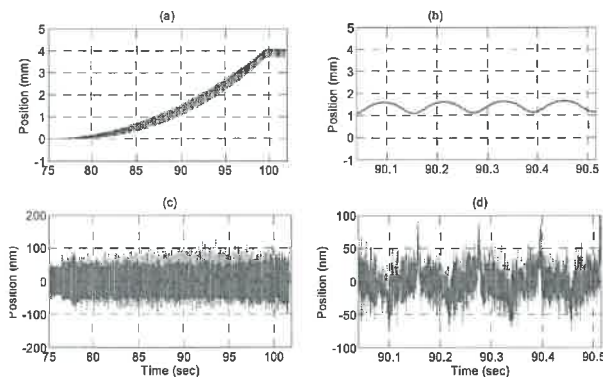


FIGURE 5. Toolpath motion for entire M4 surface as $f(\text{time})$. (a) the full toolpath; (b) close-up view for toolpath at 90 sec for two spindle revs; (c) the tool servo positioning error in full toolpath; and (d) close-up view for position error at 90 sec for two spindle revs.

Figure 5 shows a test of tool motion at the spindle speed of 500 rpm and cross-feed rate of

200 mm/min while the radial position changes from the outer radius of the optic (49 mm) to zero. The tool positioning error in the center of travel at 90 sec is less than ± 100 nm. Figure 11 also provides a close-up view of the tool motion and error magnitude for two revolution of spindle at 90 sec. The dominant motion amplitude is at the second harmonic of spindle speed because of the biconic surface.

Fiducial Surface

For the fiducial feature, the position command (Equation (2)) and acceleration command can be generated from the sinusoidal function. The polar mesh grid is created for the tool radius compensation part of surface profile as shown in Figure 6. Since the fiducial feature has an 8 mm wavelength in the circumferential direction and repeats itself every 120° , the grid covers an area larger than the size of the feature, i.e. a radial range from 58.5 mm to 64.5 mm with grid spacing of 0.1 mm and angle range of $\pm 5^\circ$ with grid spacing of 0.05° .

$$Z_{F1} = A \left(1 + \cos \left(\frac{\theta r}{8} \right) \right) \quad (2)$$

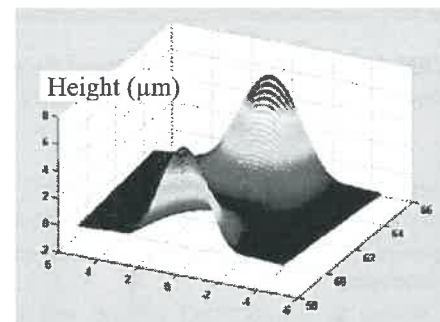


FIGURE 6. Tool radius compensation position profile for the fiducial feature

The final position command is a combination of sinusoidal position command and position correction for tool radius compensation. In addition, there are two more considerations for the tool path generation:

- (1) Because the dominant motion frequency to create the fiducial surface is 46~50 times the spindle rotational used for the optic, the spindle speed was reduced to 21 rpm. This reduction in spindle speed made the dominant frequency in the toolpath the same for both the optical and the fiducial features to produce the same phase lag.
- (2) At the start and end of sinusoidal tracking, there are larger position errors as shown in

Figure 7. This is caused by the change to the cosine motion and the acceleration feedforward controller is not as effective as the case of constant frequency. To reduce this error, only the middle wave machines the surface - position error is $\leq \pm 500$ nm.

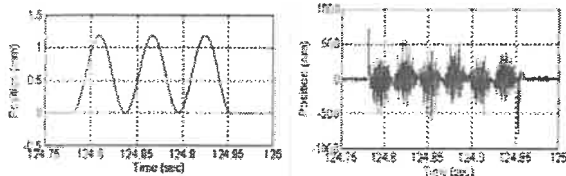


FIGURE 7. Toolpath for fiducial feature

Maching Process

Rough Machining

Conventional CNC machining was used to create an optical blank from 6061-T651 aluminum. The thickness of the blank was 100 mm to avoid any distortion of the optical surface.

Initial Diamond Machining:

A diamond tool with regular clearance angle and tool radius of 1 mm was used to machine three surfaces with following specifications

- Reference flat between the radial position 57 mm and 49 mm
- Spherical concave surface with a radius of 304.881 mm and a depth of the apex 3.943 mm below the reference flat.
- Machine top of fiducial features to be with 1.9 mm above reference flat.

Final Machining with large clearance tool:

- (1) Machine the flat reference surface (500 rpm spindle speed)
- (2) Machine the sphere (500 rpm spindle speed). The optic was removed and measured in the laser interferometer. The error was 225 nm PV and 22 nm RMS.
- (3) Machine the M4 surface (500 rpm spindle speed)
- (4) Machine the fiducial surfaces (21 rpm spindle speed))

Figure 8 shows a photograph of the machined M4 mirror with the three fiducial features distributed 120° apart rotationally to form a kinematic coupling. The mid-point for one of the features was selected as the zero degree rotationally which is coincident with the rotational reference of the M4 surface. As a result, the orientation of the optical features has been transferred to the fiducial features. The height relationship is established from the flat reference surface to both the optical shape and the fiducial

features. The shape of each of these surfaces has been measured. A CGH for the M4 surface is available and has been used to measure the optical surface and an air bearing LVDT has been used to measure several rings on the optical and fiducial surfaces to assess their relationship.



FIGURE 8. Machined M4 mirror with fiducials

CONCLUSIONS

The FLORA servo has been integrated to a DT machine and used to produce freeform optical surfaces with accompanying fiducial surfaces for assembly. The mechanical design, high-resolution encoder and high-speed control system have achieved a breakthrough in multi-element optical fabrication and assembly. A new design, FLORA II, is operational and has exhibited improved performance over the system demonstrated here.

ACKNOWLEDGEMENT

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THE MILLING DYNAMICS “SUPER DIAGRAM”: COMBINING STABILITY, SURFACE LOCATION ERROR, TOOL WEAR, AND UNCERTAINTY

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INTRODUCTION

Limitations to milling productivity include tool wear, positioning errors of the tool relative to the part, spindle error motions, fixturing concerns, programming challenges, and the process dynamics. Many research studies have been completed to address these issues. Taylor established an empirical basis for the relationships between cutting parameters and tool wear, which is still used in various forms today [1]. Other issues are detailed in texts by Trent and Wright [2] and Tlustý [3], for example.

In this work, the productivity limits imposed by the process dynamics and tool wear are considered. For a particular milling system, the tool point frequency response function (FRF) and force model enable prediction of the stability limit and location of the machined surface due to forced vibrations (surface location error, or SLE) as a function of spindle speed and cut geometry. Frequency domain solutions for the stability limit have been available for many years [e.g., 4-5]. In these analyses, the data is presented graphically in the form of stability lobes, which separate stable combinations of spindle speed and axial depth from unstable pairs. Recently, a frequency domain solution for SLE was demonstrated [6].

In this paper, the stability limit is augmented with the user selected maximum allowable SLE to produce the milling “super diagram” which identifies stable zones that also meet accuracy requirements. Additionally, the effects of tool wear are incorporated by applying wear status-dependent cutting force coefficients to the calculation of the stability limit and SLE. Finally, model and data uncertainty are included as user-specified safety margins. These safety margins capture the user’s beliefs regarding how close (in spindle speed and axial depth) he/she is willing to operate relative to the predicted limits.

SUPER DIAGRAM DESCRIPTION

The milling “super diagram” combines stability and SLE information. The predicted stability limit is set at a particular axial depth of cut for each spindle speed within the range of interest once the system dynamics, radial depth of cut, and force model are specified. The threshold for the acceptable surface location error level is user defined, however. The force model required for both stability and SLE predictions relates the milling parameters to cutting force. See Eq. 1, where F_t is the tangential force component, K_t is the tangential cutting force coefficient, b is the axial depth of cut, h is the instantaneous chip thickness, K_{te} is the tangential edge (plowing) coefficient, F_n is the normal force component, K_n is the normal cutting force coefficient, and K_{ne} is the normal edge coefficient [7].

$$\begin{aligned} F_t &= K_t b h + K_{te} b \\ F_n &= K_n b h + K_{ne} b \end{aligned} \quad (1)$$

To construct a diagram, the user selects radial depth of cut, feed per tooth, and the domain of spindle speeds and axial depths. This domain is then discretized into a grid of test points. The stability boundary is calculated using the method in [5]. Next, SLE is calculated using the approach in [6] for each grid point and compared to the user-selected limit.

After stability and SLE are calculated, penalties are applied to each of the test points as necessary. Points that are stable and within SLE tolerance levels are not penalized; their value is set to zero. Points that are stable, but outside the SLE tolerance levels are penalized by one unit (value = -1). Unstable points are penalized by two (value = -2). The point values for the selected grid are then used to construct a contour plot that serves as the super diagram. A gray-scale scheme is used to separate the different zones. The feasible operating zone (point values = 0) is white, the SLE-limited zones

(-1) are gray, and the unstable zone (-2) is black.

INFLUENCE OF TOOL WEAR

Traditionally, tool wear is treated by identifying the time it takes for a predetermined wear level (e.g., flank wear width or crater depth) to be reached for the selected machining conditions [1-3]. While this approach does specify the preferred time between tool replacements, it does not capture the corresponding changes in the process behavior. Specifically, as the tool wears, the cutting forces tend to grow. This, in turn, influences the stability limit and SLE.

To incorporate tool wear effects into the super diagram, the variation in force model coefficients with tool wear was investigated [8]. A 19 mm diameter inserted endmill (one square uncoated carbide insert with zero rake/helix angles, Kennametal 107888126 C9 JC) was used to machine 1018 steel while monitoring the cutting forces with a table-mounted force dynamometer (Kistler 9257B). The four cutting force coefficients in Eq. 1 were identified intermittently while wearing the tool by performing a linear regression to the mean x (feed) and y direction forces over a range of feed per tooth values: {0.03, 0.04, 0.05, 0.06, and 0.07} mm/tooth [7]. The feed per tooth while wearing the tool was kept constant at 0.06 mm/tooth, however. The radial and axial depths of cut were 4.8 mm (25% radial immersion) and 3 mm.

In addition to monitoring the cutting force, the insert wear status was also measured at intervals. To avoid removing the insert/tool from the spindle, a handheld microscope (60x magnification) was used to record the rake and flank surfaces. The calibrated digital images were used to identify the flank wear width (no crater wear was observed).

To demonstrate the relationship between force coefficients and wear status, the following results are provided for 2500 rpm spindle speed tests (three data sets were collected and the results were averaged). Figure 1 shows the change in K_t and K_n with the volume of material removed. Figure 2 displays the flank face images after removing 11.6 cm³ (left) and 271.6 cm³ (right) – the first and last points in Fig. 1. Figure 3 shows the corresponding variation in K_{te} and K_{ne} from the same tests.

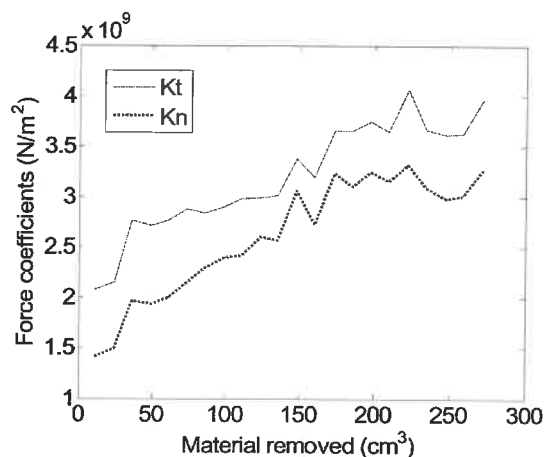


FIGURE 1. Change in K_t and K_n with volume of material removed (2500 rpm).

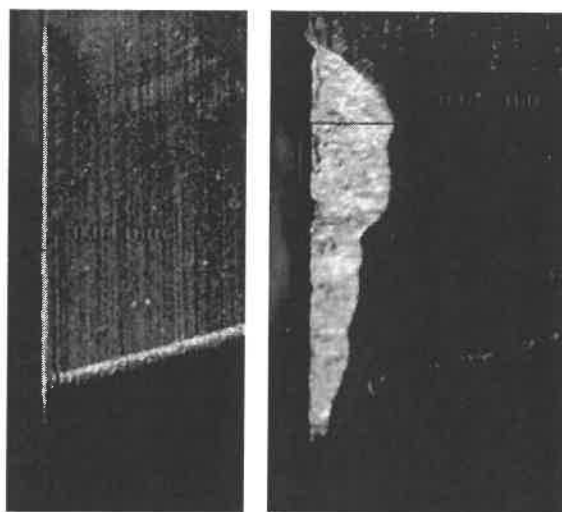


FIGURE 2. Flank wear for two testing intervals (2500 rpm).

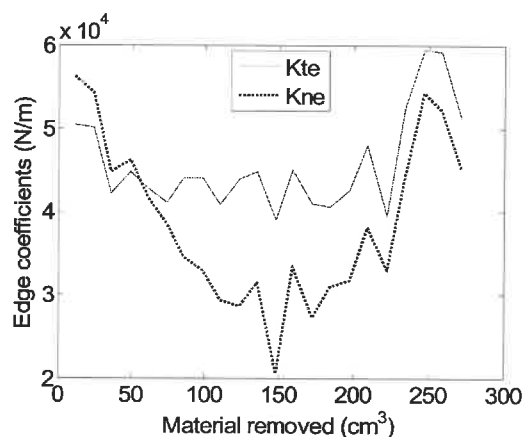


FIGURE 3. Variation in K_{te} and K_{ne} with volume of material removed (2500 rpm).

Tests were also completed at {5000 and 7500} rpm. As expected, the wear accelerated with increased cutting speed. Figure 4 shows the variation in K_t with volume of material removed for all three speeds. Similar results were observed for the other three coefficients. It is noted, however, that the relationship between force coefficients and flank wear width was not speed dependent; see Fig. 5 for the K_t behavior. Similar results were obtained for the remaining coefficients.

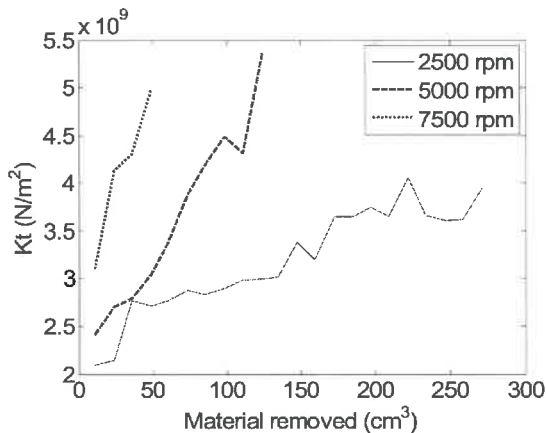


FIGURE 4. Spindle speed dependence of K_t vs. volume of material removed.

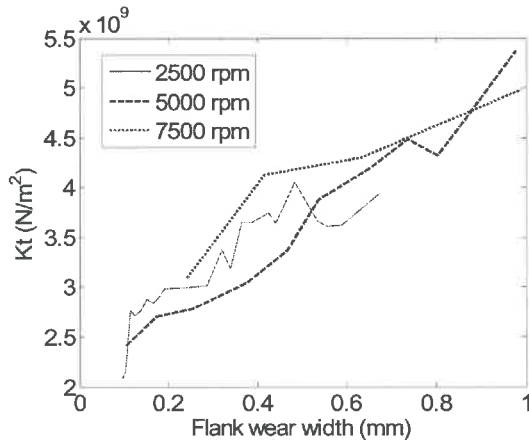


FIGURE 5. Comparison of K_t vs. flank wear width results for three spindle speeds.

INCORPORATING TOOL WEAR

To incorporate the influence of tool wear in the super diagram, the force coefficients are modified to indicate the wear status. In this study, coefficient values were chosen based on a user-selected volume of material removed. Figure 6 displays the results for a new cutting edge with the same operating conditions as previously specified. For this diagram, the x

direction tool point FRF is shown in Fig. 7 (y was similar) and the force coefficients are provided in Table 1; the SLE limit is 50 μm . Because the dominant natural frequencies for the tool point FRF occur at 3000 Hz to 5000 Hz and there is a single insert, the stability lobes and SLE variation ranges are closely spaced in the selected spindle speed range. Figure 8 shows the results after 50 cm^3 has been removed. As shown in Fig. 4, the force model coefficients are now spindle speed dependent. In other words, the stability limit and SLE predictions for the selected coefficients are only valid for one spindle speed or a limited range of spindle speeds. To address this issue, the grid of test points was separated into three regions. For spindle speeds, SS, from {1250 to 3750} rpm, the 2500 rpm coefficients were applied. Similarly, from {3750 to 6250} rpm, the 5000 rpm coefficients were used and from {6250 to 8750} rpm, the 7500 rpm coefficients were applied. See Table 1. By comparing Figs. 6 and 8, it is clear that the feasible zone is strongly influenced by the tool wear status.

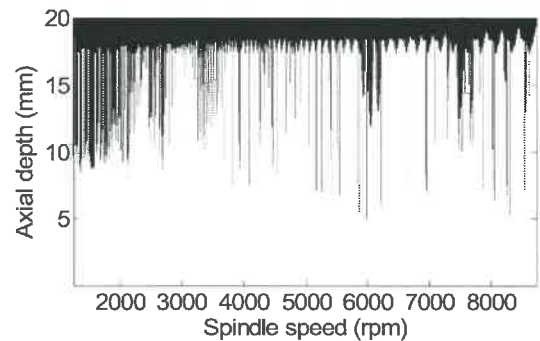


FIGURE 6. Super diagram for new tool.

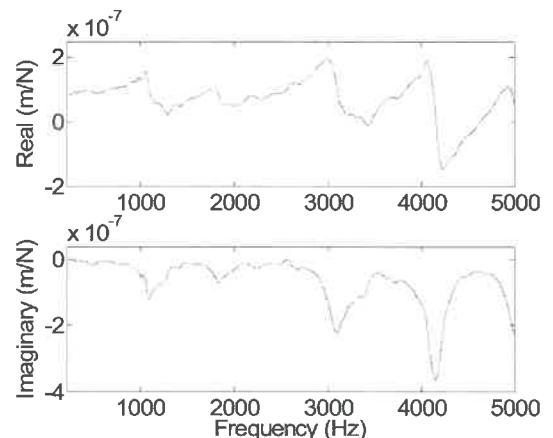


FIGURE 7. Tool point FRF (x direction).

SAFETY MARGINS

The limit between stable/unstable behavior, as well as between acceptable/unacceptable SLE values, is indicated in a binary format in Figs. 6 and 8. In practice, however, there is uncertainty in the actual location of these boundaries and they should be represented by probability distributions [9]. To incorporate uncertainty into the super diagram, user-dependent safety margins are applied to modify the feasible zone in Fig. 6. The user selects the how close in spindle speed and axial depth he/she is willing to operate relative to the predicted limits. Points within the feasible zone which violate these margins are then penalized and a new "safe" feasible zone is identified; see Fig. 9 where the safe distances from the deterministic boundaries are 100 rpm and 0.5 mm. There is now a second gray level. Dark gray indicates the stable points where the SLE limit was exceeded, while light gray represents the previously feasible points which violate the safety margins.

TABLE 1. Cutting force coefficients.

Volume removed = 0 cm ³				
SS (rpm)	K _t (N/mm ²)	K _n (N/mm ²)	K _{te} (N/mm)	K _{ne} (N/mm)
2500	2090	1420	50	56
Volume removed = 50 cm ³				
2500	2710	1940	45	46
5000	3040	2370	59	60
7500	4980	3610	92	91

ACKNOWLEDGEMENTS

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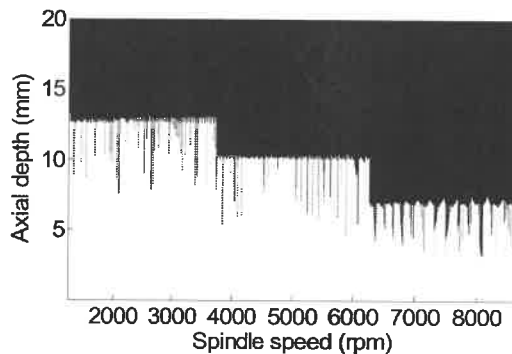


FIGURE 8. Super diagram for worn tool (50 cm³ volume removed).

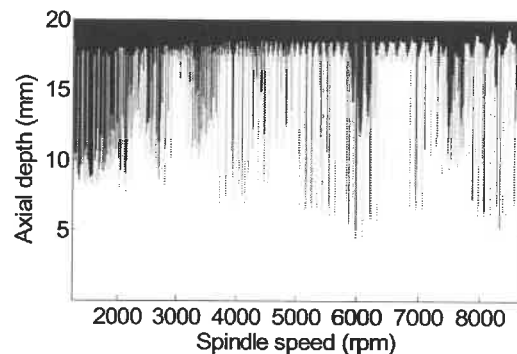


FIGURE 9. Super diagram for new tool with safety margins.

MACHINE TOOL PERFORMANCE REQUIREMENTS ASSOCIATED WITH MODULATED TOOL-PATH CHIP BREAKING

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INTRODUCTION

Turning operations involving ductile materials frequently generate long, stringy chips that present a hazard to both the machinist and the machine tool. In addition, the post-machining handling of stringy chips is a potentially dangerous and often expensive endeavor. While a variety of chip-breaking techniques are available, they generally depend on a mechanism for increasing the bending of the chip or the introduction of a one dimensional vibration that produces an interrupted cutting pattern. Unfortunately, neither of these approaches is particularly effective when making a "light depth-of-cut" on a contoured workpiece. This paper discusses a research project involving the Y-12 National Security Complex and the University of North Carolina at Charlotte in which unique, oscillatory part programs are used to continuously create an interrupted cut that generates pre-defined, user-selectable chip lengths [1, 2].

The modulated tool-path chip breaking technique overlays an oscillatory motion on top of a conventional machining tool path to periodically engage and disengage the cutting tool from the cut, as shown in figure 1 for a cylindrical workpiece. (The technique also works with contoured workpieces because the oscillation is programmed to follow the baseline tool path.) This composite tool path also produces irregular waves in the cut face, as well as complex surface-texture features.

The ability to break chips with a particular set of process parameters is dependent on the relationship between the spindle speed and the

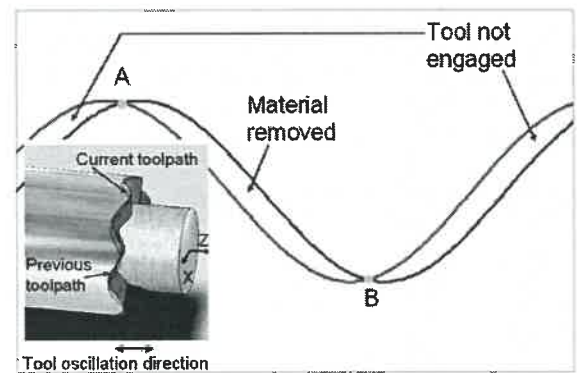


FIGURE 1. Exaggerated view of a cylindrical workpiece cut face showing the waves created by the tool-path oscillation.

oscillation frequency. It has been demonstrated that relatively small changes in spindle speed or oscillation frequency are sufficient to create the phase shift needed to produce either continuous or broken chips for a particular set of machining conditions. In addition, a model has been developed that allows the user to predetermine the process parameters needed to obtain a particular chip length, as well as define the modifications, in either the spindle speed or the oscillation frequency, needed to maintain the chip breaking action as the tool position changes with respect to the spindle centerline.

The surface texture created by a conventional turning operation creates "barber-pole" surface-texture scallops along the length of the workpiece shown in figure 1. However, these surface texture features are altered, in a complex fashion, by the modulated tool-path motion. Instead of exhibiting a "uniform thread"

characteristic, the surface texture is determined by the composite impact of three machining passes that occur in two directions with a periodically changing feedrate.

Laboratory tests have shown the viability of the modulated tool-path chip breaking technique; however, there is still a significant amount of work to be done in order to provide a "production-ready process" that takes into account the desired workpiece attributes and process throughput, as well as the necessary machine tool/control system characteristics and the part programming requirements. This paper presents the results of machine characterization activities leading to the definition of machine performance requirements associated with specific workpiece characteristics and quality parameters. Other related topics planned for investigation include cutting tool and machine wear, an evaluation of the impact on the material properties of the part, and the thermal effects/benefits associated with the interrupted cut.

EXPERIMENTAL RESULTS

A chip length prediction model has been developed that estimates the uncut chip length that will be produced for a particular combination of process parameters (oscillation amplitude and frequency, spindle speed, feedrate, and workpiece radius). This model assumes a "perfect machine" that is able to follow the tool-path commands with no errors and provides the basis for determining the process parameters that are needed to assure that sufficient phase shift exists between cut-face waves for sequential rotations of the workpiece. (The chip will be continuous if the oscillation motions are in phase with the spindle rotation.) However, machines are not perfect and it is necessary to understand the ability of a particular machine to follow a given set of tool-path modulation commands in order to obtain the desired chip characteristics.

Figure 2 shows an example of the use of capacitance sensors to characterize the ability of a machine to follow the oscillation commands as well as examine possible cross coupling motions between the axes. The sensors were aligned with the axes direction of travel and used with oscillatory part programs that moved the axes, both individually and simultaneously, through a series of frequencies that covered "zero-crossing" feedrates from 1 to 20 inches per

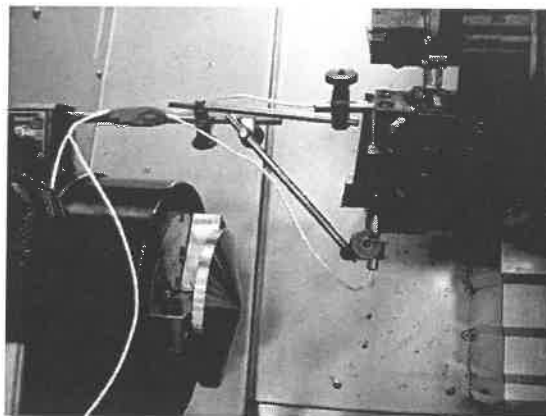


FIGURE 2. The use of capacitance sensors to measure axes motions.

minute. (Oscillation amplitudes of ± 0.005 " and ± 0.01 " were used due to the range limitation of the motion transducers.) Figure 2 also shows the tapered workpiece that was used to validate the model predictions.

Three types of oscillation motion commands were evaluated for the machine excitation testing: a saw tooth, a six-segment sinusoid, and a 10-segment sinusoid. The saw tooth excitation wave has the advantage of requiring the fewest part program blocks; however, the response of some machine tools exhibited significant amplitude degradation as the oscillation frequency was increased. These machines generally performed better with the sinusoidal excitation waveforms and the six-point option shown in figure 3 was chosen to minimize the size of the part program required to perform the tool-path modulation.

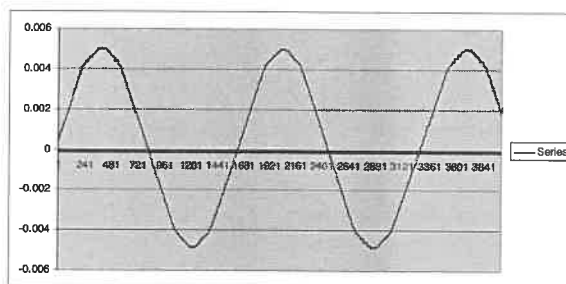


Figure 3. Response to six-point sinusoid.

Figure 4 shows the overall response of a mill-turn machine to a series of sinusoidal commands that range from 1.56 to 6.25 Hz. While the machine is able to follow the shape of the sinusoid command relatively well, the

degradation in amplitude is significant; however, the frequency response is close to what was commanded in the part program. (Prior to collecting the machine performance data, additional tests were conducted to evaluate the background vibration level as well as the frequency response of the sensor brackets.)

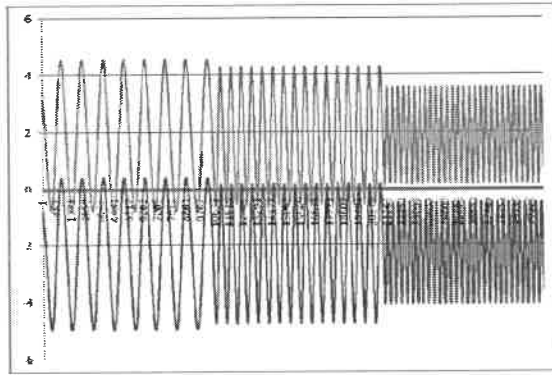


Figure 4. Machine response to sinusoidal commands of 1.56, 3.1 and 6.25 Hz. (X axis: top. Z axis: bottom.)

Figure 5 shows an expanded view of the response of a different machine to a 1.56 Hz sinusoidal Z axis position command. This machine is able to maintain an accurate peak to peak response over the range of test frequencies but it accomplishes this by degrading the frequency of the response oscillation. In addition, the shape of the response waveform is distorted by an end of block pause that occurs upon each transition to a new part program block. The X axis response shows little coupling between the axes motions.

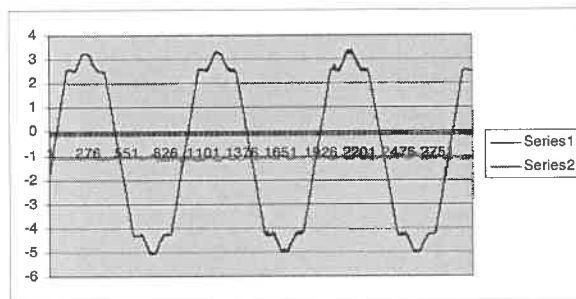


Figure 5. Expanded view of a machine's response to a 1.56 Hz. oscillation command to Z axis). (X axis response is shown just below ordinate. Vertical scale is 0.0024"/division, horizontal scale is 0.001 sec/point.)

Figure 6 shows a test set up used to characterize the machine motion errors that occur perpendicular to the tool path when both axes are given identical sinusoidal displacement commands. This test shows the cumulative effect of errors in the response of the individual axes as well as the synchronization of the axes.

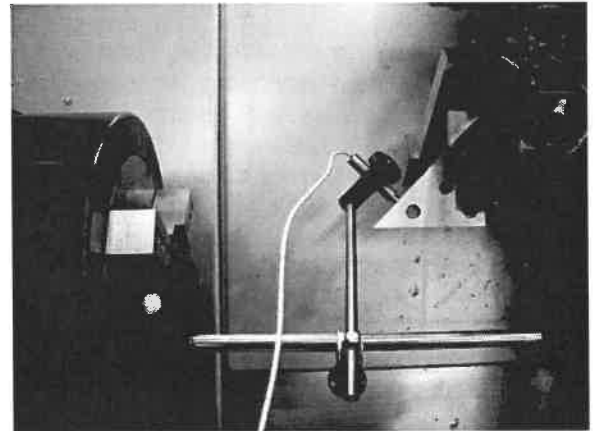


Figure 6. Test set up for measuring "out of plane" errors.

Figure 7 shows an example of the results from an out of plane error test for a precision lathe. The error motions shown are in phase with the 3.1 Hz oscillation commands, have an amplitude of approximately 0.0001", and would likely show up as a surface texture disturbance due to the relatively short wavelength of the oscillation.

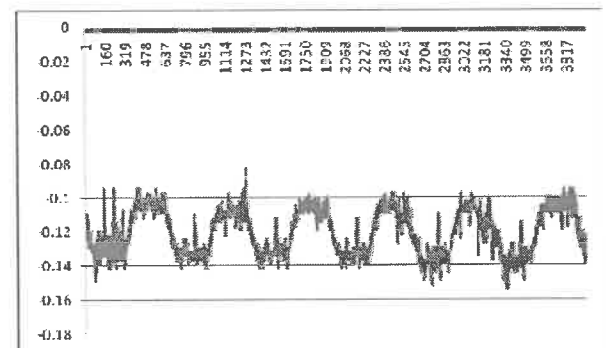


Figure 7. Precision lathe: out of plane motion errors associated with 3.1 Hz tool-path modulation commands. (Vertical scale is 0.00005"/div.)

Figure 8 shows the results of a 3.1 Hz oscillation test conducted on a less precise tool room lathe.

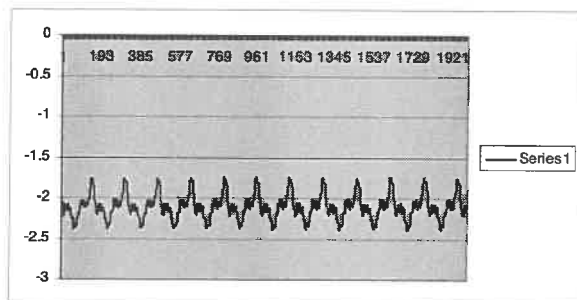


Figure 8. Tool room lathe: out of plane motion errors associated with 3.1 Hz tool-path modulation commands. (Vertical scale is 0.0013"/div.)

Figure 9 shows another technique for estimating out of plane errors. In this case, the axes responses for a single oscillation cycle, using the test arrangement shown in figure 2, are plotted against each other on the same graph. The result would be a straight line, on a 45 degree angle, if the machine's axes responded identically to the oscillation commands. Deviations from a 45 degree line are associated with errors in the position amplitude and phase.

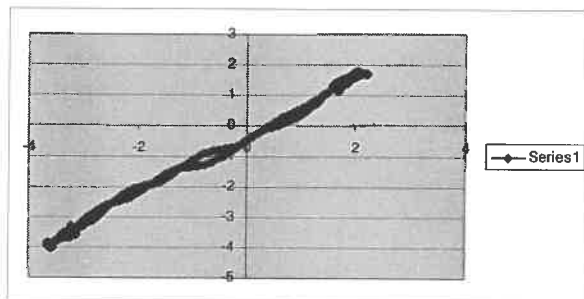


Figure 9. Comparison of X and Z axes responses over a single oscillation cycle.

FUTURE ACTIVITIES

The test results obtained to date indicate that it will be necessary to tailor the tool path modulation parameters to a specific machine's performance characteristics. Depending on production needs, this may mean selecting machining process parameters based on potentially conflicting requirements such as throughput, chip length, workpiece quality parameters, etc. A key element in performing the process parameter selection task is a reliable model that allows a user to determine the impact of various combinations of parameters. Preliminary machining tests, using the test part shown in figure 2, indicate that the parameter selection process can be very non-

intuitive and it may be desirable to use different parameter values for roughing and finishing operations. In addition, other factors such as tool temperature and tool wear may be important factors in some applications and will need to be incorporated into the modeling process.

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THE INFLUENCE OF MACHINING PARAMETERS AND TOOL GEOMETRY ON THE QUALITY OF INFORMATION CARRYING MICROSTRUCTURES

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INTRODUCTION

Within the Collaborative Research Centre (CRC) 653, components with novel properties as well as concepts, methods and techniques for their manufacturing and use in production engineering are developed [1]. The CRC's long-term objective is the integration of associated information on the very same component, like the date and place of assembly or the part numbers. Therefore, these information have to be intrinsically tied to the component and should be readable on demand. This paper presents an approach for the linking of rotation-symmetric components with corresponding information by means of surface microstructures machined by a piezo-electrically actuated tool named "StrucTool" (ST). This tool concept is characterized by high tool acceleration and is either applied in the machining of precision contoured and structured optical surfaces [2, 3, 4] or for measurement purposes [5].

To use the component itself for information storage, defined microstructures are produced at its surface by an external turning process. The lateral movement of the tool is superimposed by a highly dynamic oscillation of the tip in the radial direction of the workpiece which results in a varying but though continuous cutting depth. Cutting speed and feed interact with the tool tip movement and thus create microstructures. The procedure is illustrated in FIGURE 1. Micro structures for data storage can e. g. be applied on shafts or axes. However, the microstructured areas must not be subject to thermal or mechanical load. Therefore, especially those shaft sections which are not in contact with other components either in their actual use or in the assembly are most suitable.

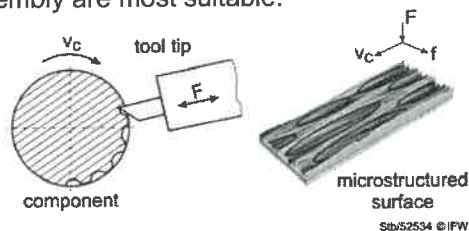


FIGURE 1. The principle of micro structuring.

DATA STORAGE PROCESS

In order to tie information intrinsically to the component surface, a data storage system has been developed which is based on the principle of data transmission in telecommunications engineering. The innovation consists in the "freezing" of the transfer signal on the component surface. The signal is introduced into the surface by defined microstructures and is thus stored. On demand, the data can be read out by means of an optical procedure ([5], FIGURE 2).

First, the necessary data are translated into ASCII (Source Encoding). Afterwards this ASCII code will be adapted to work appropriate with the following transmission path (Channel Encoding). Then the data set is transferred into several sinusoidal oscillations with varying frequencies (Modulation). In the turning process, the ST is actuated by means of this signal and introduces the information into the component surface. When the microstructures are read-out in order to reclaim the stored information the described modulation and encoding procedure is done vice versa.

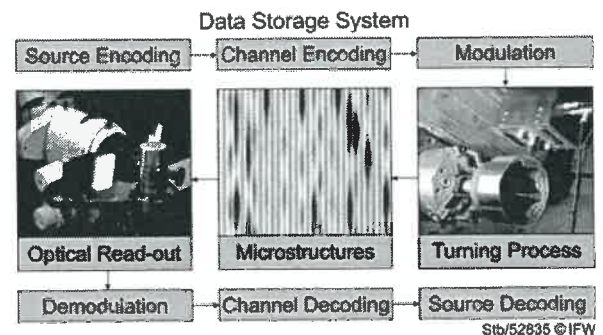


FIGURE 2. Data Storage System.

StrucTool

The piezo-electrically actuated StrucTool (FIGURE 3) has been developed regarding the presented application. In the external turning process, this tool overlays the amplitude with the cutting speed and the feed speed. Thus a structure geometry can be produced by a

variation in the cutting speed, the amplitude and the frequency. The ST is integrated into a lathe of the type Gildemeister TWIN32 by means of a VDI3245 holder.

Due to the stimulation of piezoelectric stack actuators which are supported by the bottom of the housing, an extension of the solid actuator is directly transferred by the flexure hinge into a motion of the indexable insert. The tool amplitude can on one hand be adjusted by connecting the actuators either in series or in parallel. On the other hand it is possible to scale the actuator controlling current by the amplification factor K . The actual position of the tool tip is surveyed by the built-in position transducer.

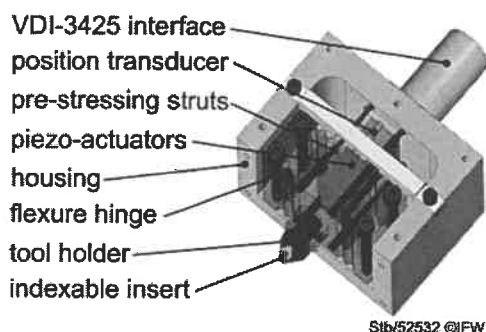


FIGURE 3. Piezo-electrically actuated tool.

To determine the transfer function of the ST its amplitude has been recorded using a laser vibrometer while the tool was actuated by means of a chirp signal. FIGURE 4 shows the surveyed amplitude response depending on the applied frequency without tool engagement. The first natural frequency of the tool is of about 6500 Hz and it is independent of the amplification factor K .

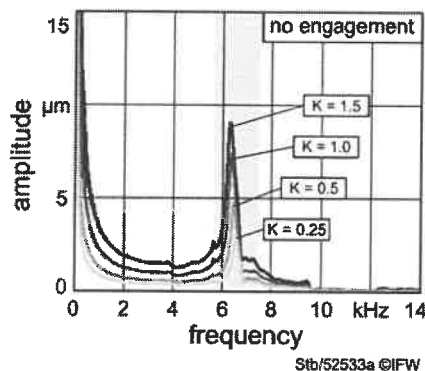


FIGURE 4. Amplitude response of the ST.

In order to avoid phase displacement of the signal within the range of the natural frequency (FIGURE 5), all examinations concerning the microstructuring process are performed between 2000 Hz and 5500 Hz. 2000 Hz have been chosen as the lower limit to provide a minimum of data density while storing information.

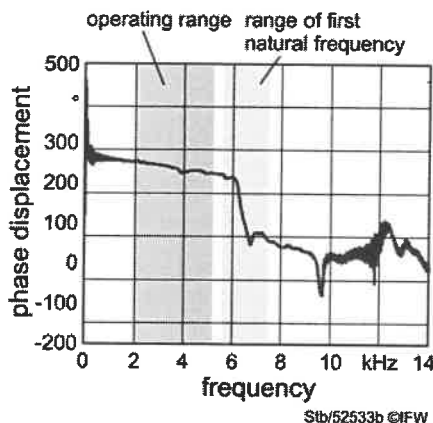


FIGURE 5. Phase displacement of the ST.

When the tool cutting edge is engaged with the working material, the damping of the amplitude response is nearly constant up to the natural frequency. Thus it can be assumed that the displacement of the tool tip is reduced constantly by 25 percent of the maximum amplitude at different frequencies within the operated frequency range and with varying amplification factors.

INFLUENCE OF PROCESS PARAMETERS

For the examinations, AISI1045 steel has been applied, because there is much knowledge available on the machinability of this material. Preexaminations have shown that the tool tip geometry is decisive for the quality of defined microstructures [6]. In this case the term "quality" describes the applicability of the microstructures for the storage of information on a component regarding waviness and roughness. In order to determine the relationship between the suitable tool geometry and the structure quality, the tool corner radius r_c and the tool rake angle γ have been varied systematically. Concerning the actuating variables of the ST, the structuring frequency F and the amplification factor K are examined; with regard to the turning process, cutting speed v_c , feed f and the depth of cut a_p are examined. The clearance angle α and the tool corner angle ϵ are kept constant at 7° resp. 80° in all

examinations. The parameters which have been varied within the investigations are listed in TABLE 1. The specified default values are kept constant while the one which is examined is varied.

TABLE 1. Varied parameters.

Parameters	Abbr.	Default Values	Variation
Cutting Speed	v_c	250 m/min	50...310 m/min
Infeed	f	70 μm	20...90 μm
Depth of Cut	a_p	100 μm	25...100 μm
Corner Radius	r_c	100 μm	40...400 μm
Frequency	F	2000 Hz	2000...5500 Hz
Amplification Factor	K	1.5	0.25...1.5

For the evaluation of the process quality, the waviness w_t^* and the roughness Ra^* in cutting direction are analyzed. In this case, the surface evaluation follows DIN EN ISO 4287 despite that the parameters have been measured in cutting direction and by a non-contact confocal measurement system.

Machining Parameters

The influence of the cutting speed on structure quality is illustrated in FIGURE 6. On the left hand side the correlation of the roughness Ra^* and waviness w_t^* with the varied cutting speed v_c is shown. To get a more detailed impression of the structured surface a grey-scaled height map is displayed on the right.

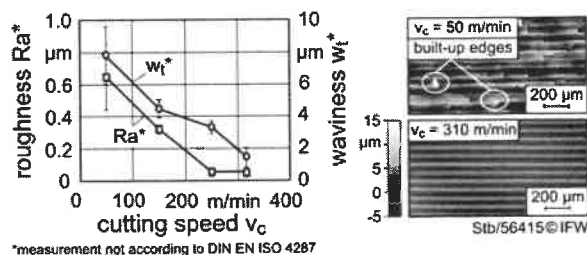


FIGURE 6. Influence of the cutting speed on the surface quality.

An increase in the cutting speed leads to an improved structure quality; both the waviness and the roughness in the cutting direction decrease. The surface images show that the quality decreases clearly at cutting speeds below $v_c = 150$ m/min. The reason for this is the formation of built-up edges [7]. At low cutting speeds, the material is piled up in front of the cutting edge and at the same time plastically deformed. Then cold work hardening takes place and the deformed material is loosened

intermittently. This can be observed by sudden single vertical heights along the cutting path. At cutting speeds above $v_c = 200$ m/min, built-up edges can not be observed.

The cutting speed also has a significant influence on the waviness. At low cutting speeds, the material break outs and the built-up material lead to high waviness. With an increasing cutting speed, the waviness is reduced due to the decrease in built-up material. Because of the measured amplitude response (FIGURE 4), a waviness of about 3-4 μm in cutting direction is expected. At $v_c = 310$ m/min, the waviness is considerably lower due to dynamic effects.

Furthermore, the influence of the feed f on the surface quality has been investigated. In order to attain a high data density, the feed has to be as low as possible. The feed does not have a significant influence on neither waviness nor roughness nor burr formation.

Also the depth of cut a_p does not have a significant influence on the surface quality as long as minimum chip thickness is not under-run.

Tool Geometry

The tool corner radius has a small influence on both the waviness and the roughness. An increased corner radius leads to an increase in the roughness Ra^* (FIGURE 7). However it has to be taken into account that the standard deviation also rises. Furthermore burr formation decreases with higher corner radii. This effect is due to the decreasing actual inclination angle with larger corner radii [8].

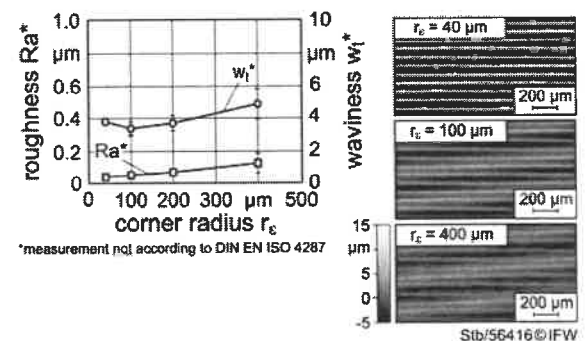


FIGURE 7. Influence of the corner radius on the surface quality.

Additionally, the influence of the rake angle γ on the structure quality has been investigated. Similar to the results observed for the feed and the depth of cut, no influence of the rake angle

on waviness and roughness was observed within the chosen range of parameters.

StrucTool Parameters

With an increasing amplification factor, the roughness at the bottom of the structure remains constant, while its waviness increases almost exponentially. The increase in the waviness is caused by the increased tool amplitude with a higher amplification factor (FIGURE 8). However this effect only occurs above amplification factor of $K = 1$. Below this value, the damping due to the tool engagement is predominant.

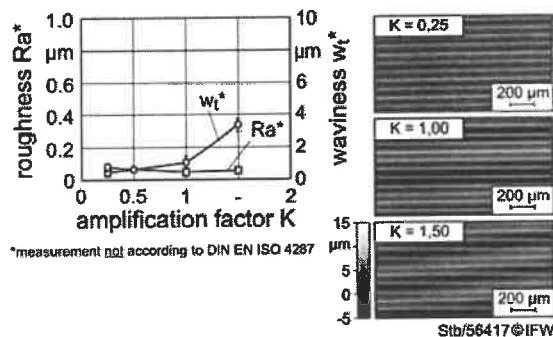


FIGURE 8. Influence of the amplification factor on the surface quality.

In further examinations, the influence of the excitation frequency F of the tool system on the structure quality has been investigated. This parameter does, in the chosen parameter range, not have any influence on the waviness or roughness of the structures either. However, the information density can be increased by higher frequency values.

SUMMARY

The presented examinations clarify that not all of the process parameters have an influence on the surface quality. Only the cutting speed, the tool corner radius and the amplification factor show a major effect. To store and read out information not only roughness and waviness have to be taken into account, but also the burr formation on the structure's ridge. This burr formation can be influenced by the corner radius of the tool tip as shown. If these burrs occur, the optical read-out of the stored data will be affected negatively.

To increase the amount of stored data low cutting speeds, low feeds and small corner radii are necessary. But due to the presented technological limitations it is not possible to increase the data density infinitely. It is

especially restricted by the cutting speed and the feed. For these parameters, certain limit values have to be observed in order to guarantee favorable chip formation without damaging of the component surface. By using the default values for process parameters a data density of about 4 kBit/cm² can be achieved.

Further investigations will deal with the transfer of the process to generate information carrying microstructures onto planar surfaces. This will be achieved by using a spindle with a piezo-electrically actuated infeed axis in a milling operation.

ACKNOWLEDGEMENTS

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METALLOGRAPHIC ORIENTATION EFFECTS ON BURR FORMATION IN MACHINING OXYGEN-FREE-COPPER USING SINGLE CRYSTALLINE DIAMOND MICRO-TOOLS

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ABSTRACT

A study was carried out to investigate the mechanism of burr formation in micro-scale-mechanical machining (henceforth referred to as 'micro-machining') of Oxygen-Free-Copper (OFC) using a 5-axis ultra-precision machine. The single crystalline diamond (SCD) micro-tools with a cutting contact length of around 30 μm on the primary clearance face were employed in this study. Burrs could be observed when the cutting depth is deeper than 0.9 μm . The crystallographic orientation exerts a great influence on the formation of the burrs. Modifying the machining parameters, such as reducing the cutting depth or the cross-feed rate could minimize the burr size. This study contributes to the understanding of the physics of mechanical machining with micro-tools.

Key words: Micro-size diamond tool, micro-machining, FIB, grain orientation, burr formation

INTRODUCTION

Micro-burrs have been observed in the micro-milling of stainless steel, brass, aluminium and cast iron [1, 2]. The suppression of burr development in machining with a micro-tool is very important because, unlike in macro-machining, post-processing cannot always be applied to remove burrs on miniature fabricated parts. De-burring may introduce dimensional errors and residual stresses in the component. Therefore, the best solution is to prevent or minimize burr formation during micro-machining. For the implementation of this approach it is critical to understand the basic mechanisms involved in the burr formation in micro-machining using micro-tools. However, since the metallographic phase size is often of the same order of magnitude as the cutter size, the metallurgical structures will affect the overall

cutting properties, leading to a distinct difference between the traditional macro-mechanical machining and the micro-machining. A number of studies have been carried out to arrive at a better understanding of the effects of metallographic structure on machining single crystal materials [3-9]. Polycrystalline OFC is widely used for optical components since optical grade surfaces can be produced with diamond turning. However, so far, there is very little reported information on the introduction of burr formation as well as the minimization of burrs in micro-machining OFC. There fore, there is a needed to carry out a study to investigate the fundamental mechanisms in micro-burr formation for the purpose of predicting and controlling the burr formation during micro-machining OFC

MACHINING EXPERIMENTS

All experiments were conducted using a Moore Nanotech 350 ultra-precision Freeform Generator. A FEI dual beam focused ion beam (FIB) system (Nova NaviLab) which integrates ion and electron beams for FIB and SEM functionality in one machine was used to fabricate the micro-size SCD tools and observe the machined work pieces. A Wyko NT3300 optical profiler was used to measure the machined surface after experiments. Kistler dynamometer (9256C1 Minidyn Multicomponent Dynamometer) was used to detect the force signals during machining. A Nano-indenter (NanoTest 550) was used to test the micro hardness on the top surface of the work material.

OFC is a high purity copper with less than 0.05% residual deoxidants. Table 1 show the material's properties. Prior to micro-machining and nano-indentation, an SCD tool with nose radius of 1.0 mm was used to shape the top

surface of work materials. The shaped top surface of the work material can achieve an optical quality finish with R_a as low as 7nm and flat waviness of 0.1 μm Peak-Valley value.

The SCD micro-tool was fabricated using a FIB [10] to obtain the following profile: cutting edge radius ranging from 15 μm to 18 μm , rake angle of -1.5° , and primary and side clearance angles of 7° . The cutting contact length at the tool-workpiece interface is around 30 μm . Since machining is carried out at small depth-of-cuts using a micro-size tool, it was necessary to align the tool carefully in order to achieve high alignment accuracy.

RESULTS AND ANALYSIS

Grooves were formed on the top surface of the work material with the groove width equivalent to the cutting contact length at a tool-workpiece interface of around 30 μm and a cutting speed of 1.0mm/min. Deformation or pile-up could be observed along the groove edge when the cutting depth was deeper than 0.9 μm with the existence of burrs between A and B as shown in Fig. 1 (a). The burr was generated at the crystallographic grain right to the grain boundary and can be observed in the close-up view in Fig. 1 (b). Previous research undertaken by the authors concluded that, as the micro-size tool traverses within a grain with a higher hardness, the work material in front of the machining tool may deform severely, leading to thicker chip, striation at the chip back, higher machining force, reduced shear angle and a degraded machined surface [11]. A similar phenomenon was displayed in Fig. 1 and Fig. 2, such as the build-up at the chip, degraded machined surface and a significant increased cutting force at the instance when the burr was formed during micro-machining. The micro-hardness test indicated that the metallographic phase between A and B is harder (2.6 GPa) than that obtained in the grain to the left of the boundary (1.7 GPa). Thus when the micro-tool traverses within a grain of relative higher hardness the likelihood of burr formation is increased.

The formation of the burrs attributed to the crystallographic orientation could be avoided by modifying the machining parameters. Reducing the cutting depth helps to minimize the burr size. Fig. 3 shows that there were no burrs observed

with a variation in grain orientation at a cutting depth of less than 0.35 μm at the same cutting speed without coolant, although the machined surface changed with the crystallographic orientation.

Three grooves were formed through the same grain (upper dashed curve) by a micro-tool as shown in Fig. 4 (a). The depths of grooves from the 1st to 3rd groove were 0.9 μm , 1.4 μm and 1.7 μm respectively. The 1st and 2nd grooves along with Zone Z_1 at the 3rd groove were formed using a micro-tool with the top layer being removed in one pass and the layer width equalling that of the cutting contact length at the primary clearance face using a machining speed of 1.0mm/min. The Zone Z_2 in the 3rd groove was generated by a series of passes with the same machining speed and direction; however a cross-feed of 1 μm /pass was used resulting in material on the right side of the tool being repeatedly removed. No coolant was used in the micro-machining experiments. Burrs could be found at all groove edges except the right edge of the 3rd groove which was generated with a cross-feed of 1 μm /pass. Fig. 5 show the forces obtained from forming the right edge of the 3rd groove with a cross-feed of 1 μm /pass. Comparing to the significant change in cutting forces displayed in Fig. 2, the invariant cutting forces were observed though the micro-size tool cut as it crossed various phases with the full cutting distance of 2.4 mm at a reduced cross-feed. The reduced cutting force in micro-machining plays an important role in minimizing the burr size.

CONCLUSIONS

Burrs could be observed when the cutting depth is deeper than 0.9 μm . The burrs may form if the pile-ups or cracks/fractures occurred at the groove edges once the localized stresses builds up to a threshold when the cutting depth is deeper than a specific value. The crystallographic orientation exerts an influence on the formation of the burrs. Modifying the machining parameters, such as reducing the cutting depth or the cross-feed could minimize the burr size.

TABLE 1. Mechanical properties of OFC

Physical properties		Mechanical properties 20°C			
Density (g/cm ³)	Young's Modulus [GPa]	Ultimate Tensile Strength (MPa)	Yield Tensile Strength (MPa)	Elongation (%)	Hardness (HB)
8.94	117	331	303	16	93

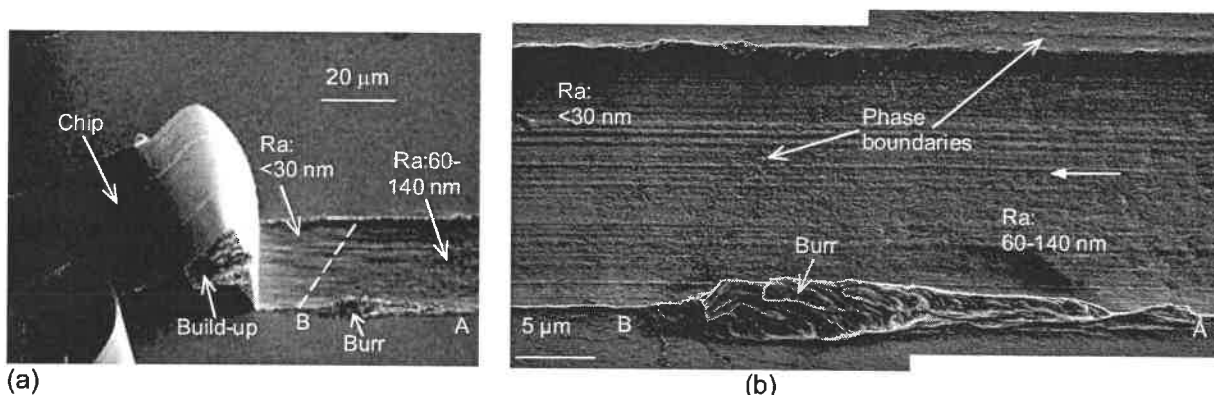


FIGURE 1. Grain orientation effects on the burrs formation in micro-machining. (a) Overview; (b) Close-up view.

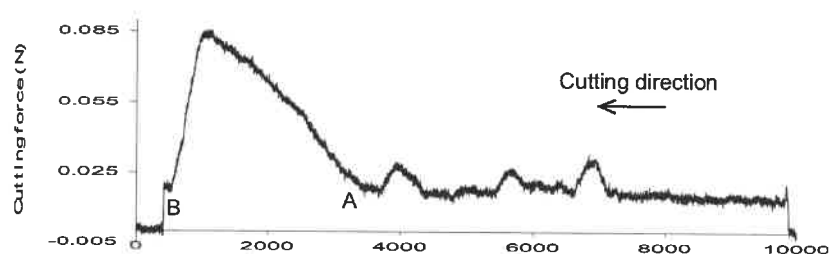


FIGURE 2. Cutting force obtained during micro-machining.

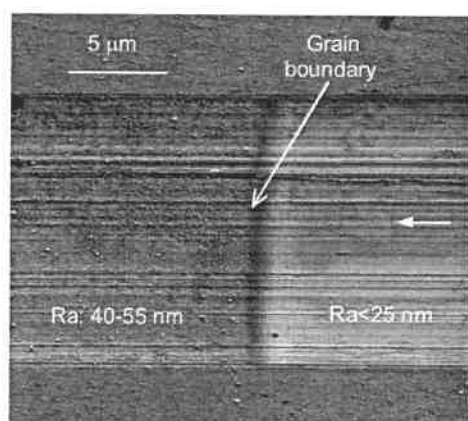
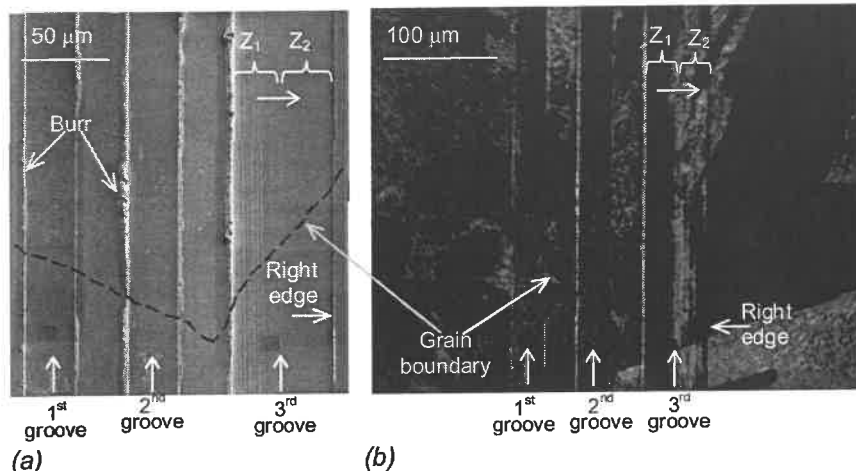


FIGURE 3. Micro-grooves machined at a cutting speed of 1 mm/min and at a cutting depth of 0.35 μ m without coolant.



(a) (b)
FIGURE 4. Three micro-grooves formed within one metallographic phase by a commercial micro-size SCD tool. (a) Scanning Electron Microscope image; (b) Scanning Ion Microscope image

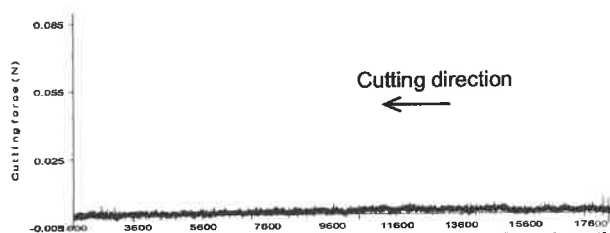


FIGURE 5. The cutting force obtained from forming the right edge of the 3rd groove.

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RECONFIGURABLE ASSEMBLY STATION FOR PRECISION MANUFACTURE OF NUCLEAR FUSION IGNITION TARGETS

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INTRODUCTION

This paper explores the design and testing of a reconfigurable assembly station developed for assembling the inertial confinement nuclear fusion ignition targets that will be fielded in the National Ignition Facility (NIF) laser [1]. The assembly station, referred to as the Flexible Final Assembly Machine (FlexFAM) and shown in Figure 1, is a companion system to the earlier Final Assembly Machine (FAM) [2]. Both machines consist of a manipulator system integrated with an optical coordinate measuring machine (OCMM). The manipulator system has six groups of stacked axis used to manipulate the millimeter-sized target components with sub-micron precision, and utilizes the same force and torque feedback sensing as the FAM. Real-time dimensional metrology is provided by the OCMM's vision system and through-the-lens (TTL) laser-based height measuring probe. The manually actuated manipulator system of the FlexFAM provides a total of thirty degrees-of-freedom to the target components being assembled predominantly in a cubic centimeter work zone.

MOTIVATION FOR DEVELOPING THE FLEXIBLE FINAL ASSEMBLY MACHINE

The National Ignition Campaign [3] goal of achieving thermonuclear fusion burn and gain with the NIF laser requires building at least one inertial confinement fusion (ICF) target per day. Referring to figures 2 and 3, the target is designed so that the physics package (hohlraum, capsule and the gas between them) can be tailored independently of the thermal-mechanical package that holds it (TMP-halves, diagnostic band, and silicon cooling arms). The required position accuracy of the assembled target components is in the range of $\pm 2\text{--}20\text{ }\mu\text{m}$. The FlexFAM has already significantly increased the production rate of NIC targets, and its

manipulator system is easily reconfigured to accommodate the different target configurations envisioned for NIC experiments.

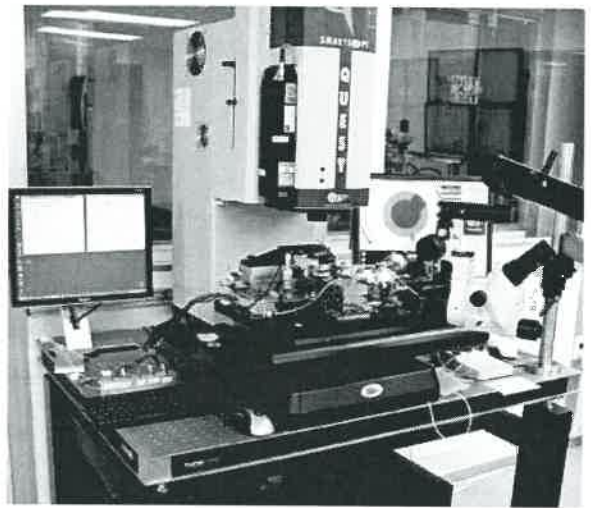


FIGURE 1. Flexible Final Assembly Machine.

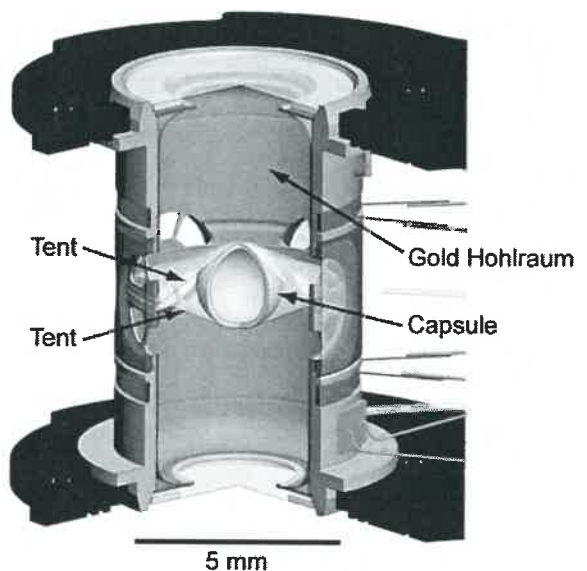


FIGURE 2. Model of a NIC target.

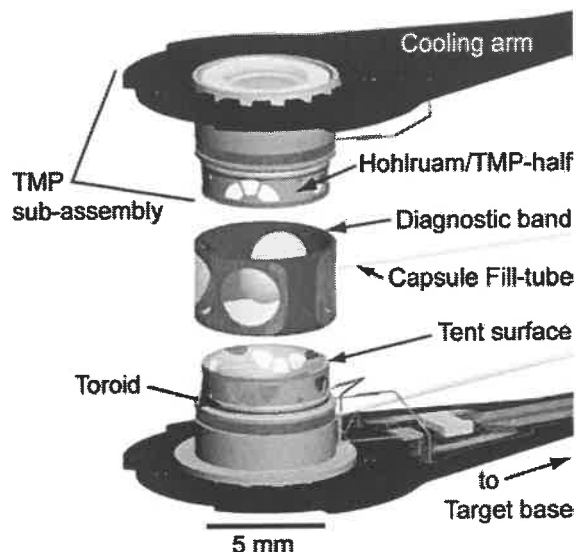


FIGURE 3. Model showing final assembly step for a target.

THE NEED FOR FLEXIBILITY

Referring to Figure 4, the manipulator system of the FlexFAM resides on a platform that can be readily removed and replaced from the OCMM. This allows configuring different manipulator systems to suit various NIC target configurations, many of which would be incompatible with one particular arrangement of manipulator stages. This flexibility reduces the down-time of the target production facility when a different target configuration is introduced. Instead of tearing down and reconfiguring a manipulator system that was arranged and precisely aligned for an earlier target configuration, it can be set aside and replaced with a different manipulator system that could be preconfigured off-line ahead of time. Targets that share similar configurations would then benefit from the rapid start-up time provided by a previously configured, aligned, and warehoused manipulator system. Moreover, the manipulator system could be removed so that the OCMM could be made available more broadly to the target fabrication facility to support dimensional metrology of target components, target sub-assemblies, and completed target assemblies.

OVERVIEW OF THE DESIGN

The FlexFAM was designed to operate in a class 1000 clean room, and consists of an LLNL-developed manipulator system integrated with a commercial OCMM having micron-level



FIGURE 4. FlexFAM manipulator system.

accuracy over a working volume of 30 cm x 15 cm x 20 cm. The manipulator system consists entirely of manually-actuated stages, which enabled a quick path to developing a thirty degrees-of-freedom motion system that is more compact and simpler to operate than the earlier FAM [2]. This choice not only allowed meeting the aggressive timeline for the project, but the smaller envelope of the FlexFAM manipulator system allowed integrating it with a smaller bench-top styled OCMM than the FAM uses, providing easier operator access to the target assembly work zone. As shown in Figure 5, all of the stages are within comfortable reach of the operator so the need for motorized stages was greatly reduced. Functional improvements to the FlexFAM were identified during the design phase by selecting certain axes that would benefit from a future upgrade to motorized stages to enhance the ergonomics and performance for the manipulator system.



FIGURE 5. Target Assembly Technician operating the FlexFAM.

As the target components are aligned and fitted together, sensors embedded in the manipulator system provide 100-milligram resolution force and gram-millimeter resolution torque feedback of the contact loads. The OCMM has a machine-vision system and a laser-based height-measuring probe that provide micrometer-accuracy dimensional measurements of the target while assembling it. The vision system and laser probe of the OCMM are used to guide the initial approach and alignment of the target components, and to measure the relative position and orientation of the components. The force and torque feedback is used to guide the final approach, alignment, and mating of delicate target components that fit together with micrometer-level or no clearance. Figures 6 and 7 provide examples of the vision system and force and torque feedback presented to the operator.

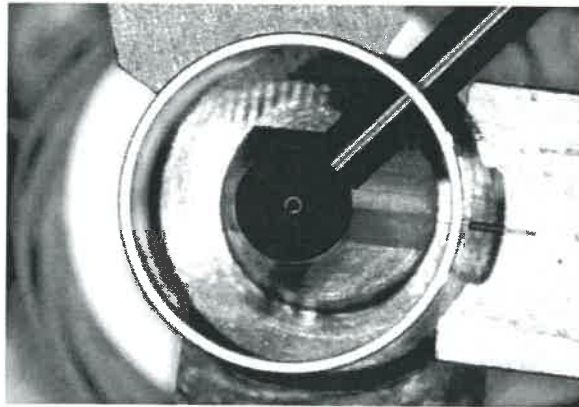


FIGURE 6. Image from the OCMM vision system showing the capsule held by a vacuum wand and the diagnostic band held by a vacuum chuck.

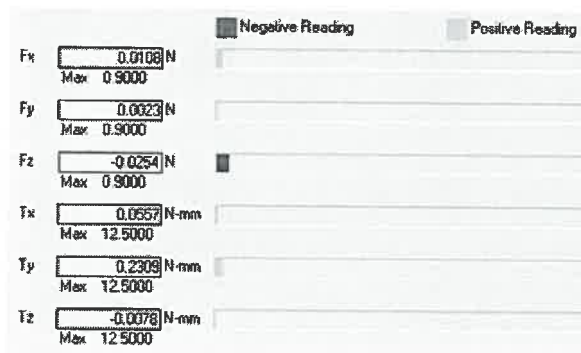


FIGURE 7. Image from the user interface for the force and torque sensor for the lower target half.

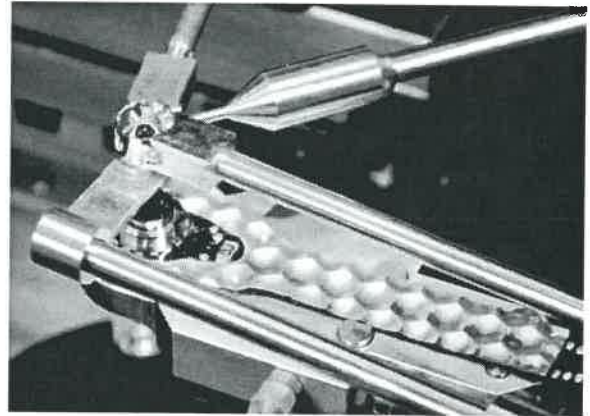


FIGURE 8. A view of the operating arena of the FlexFAM. The lower target half subassembly lies below the tooling used to manipulate the capsule fill-tube assembly and diagnostic band.

Target components are held by LLNL-developed vacuum tooling and specialized fixtures, like the ones shown in Figures 8, 9, and 10. In figure 8, the capsule is being centered in the diagnostic band. Figure 9 shows the target components aligned and ready for the final assembly step: the upper and lower target halves are brought together inside the diagnostic band so that the tent on the end of each target half closes on the capsule, suspending it in the mid-plane of the target assembly. Referring to Figure 10, incorporated in the FlexFAM tooling are kinematic and semi-kinematic mounts that ensure accurate and repeatable remove-and-replace orientation of the tooling.

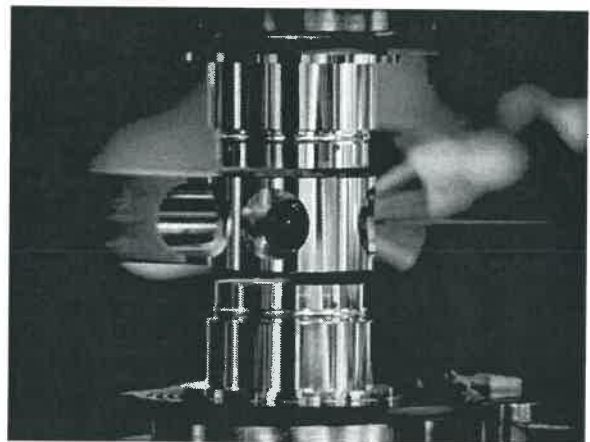


FIGURE 9. A second view of the operating arena of the FlexFAM. The capsule is centered inside the diagnostic band and between the upper and lower target halves. The capsule fill tube is visible on the right side of the figure.

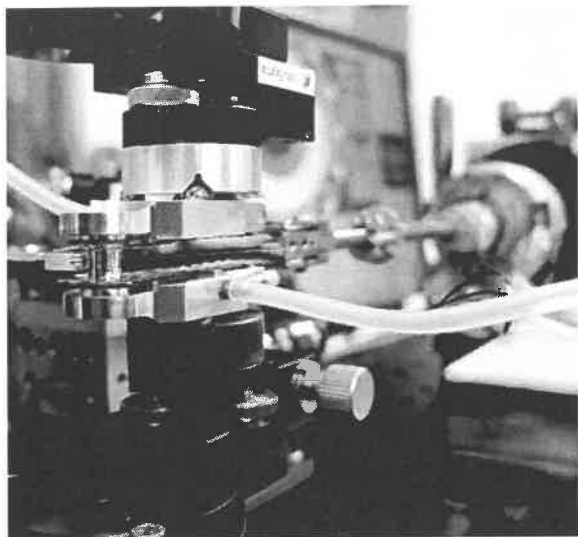


FIGURE 10. The upper and lower target half subassemblies are held in the FlexFAM by magnetically-coupled kinematic tooling mounts. The relatively long target base assembly, which gets connected to the ends of the upper and lower target half subassemblies, fades out of focus on the right side of the figure.

Future Work

The manipulator system is attached to the OCMM with fasteners threaded into the manufacturer-provided holes in the OCMM stage. A concept for a kinematic mounting system is being developed into hardware that will provide accurate and repeatable mounting and removal of the manipulator system. This will better support a major goal that the FlexFAM provide a rapid turnaround in re-tooling the target production facility in response to changes in the configuration of targets required by the National Ignition Campaign.

The novel tooling shown in Figure 8 for manipulating the capsule fill-tube assembly and diagnostic band is being migrated to the earlier Final Assembly Machine, which is also building NIC targets [2]. The specialized fixture that is used to thread the capsule fill-tube assembly into the diagnostic band is being perfected for use with both target assembly machines. As experience using the FlexFAM and its copies is gained by target production personnel, the need to motorize certain axes will be revisited.

Conclusion

The Flexible Final Assembly machine is being used to assemble production ignition targets for the National Ignition Campaign. Figure 10 shows one of these targets being built on the FlexFAM by one person in a single 8-hour shift, five months after starting this project. The FlexFAM successfully demonstrated that it can assemble inertial confinement fusion ignition targets that meet the required micron-level tolerances. Its manipulator system provides the precise and repeatable motion needed for assembling a target, the force and torque feedback allows an operator to deterministically engage the components being assembled, and the OCMM provides the requisite alignment of components during the assembly process and provides an immediate verification of the accuracy of the assembled target. The simplicity of use afforded by the manually-actuated manipulator system is allowing a rapid transfer of the FlexFAM from its development team to the target production team. Novel target assembly tooling developed for and tested on the FlexFAM is being migrated to other machines being used to build NIC targets.

Acknowledgment

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FABRICATION OF A PRECISION ENGINEERED MATERIAL BY MICRO-WIRE EDM

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INTRODUCTION

Sandia National Laboratories' Meso Manufacturing group was approached by a customer with a need for a high density, engineered material that could be tuned to fail at specified mechanical shock loadings. The challenge of creating an engineered material is significant. The material desired for this application is one that has tightly tuned shear fracture loading level in 2 axes, with high compressive strength in the 3rd axis. It is also desirable that the material be extremely dense. The selected material also needed to have a compatible manufacturing process which would be capable of creating features measuring 10's of micrometers without imparting significant force to the part or to the delicate features.

CREATING AN ENGINEERED MATERIAL

The team first investigated existing materials, but found none that satisfied all of the requirements. Then the team investigated numerous methods of manufacturing an engineered material. Some methods offered precise features, while others offered high density material. The chosen solution promised to fulfill all of the requirements using a proven manufacturing method. The chosen solution is to create a material by utilizing tungsten as the dense and brittle material coupled with micro-wire electro discharge machining (μ WEDM). The requirements for tuned shear strength are achieved by cutting the surface of the tungsten into cube-shaped blocks positioned on top of small "blades" of thin cross-section. Conceptual drawings of this block-on-blade structure is shown individually and in an array in FIGURE 1.



FIGURE 1. Conceptual models of engineered material show the block on blade structure individually (left) and in an array (right).

Tungsten was selected as the material to be cut by μ WEDM because of its high density, its electrical conductivity (necessary for EDM) and its relative brittleness when compared with other highly conductive, dense materials (primarily metals). Early investigations into the properties of commercially available tungsten revealed several challenges. The first is that the properties of tungsten vary greatly depending on the manufacturer and the manufacturing method. Most tungsten is made by powder metallurgy in which tungsten powder is pressed into a form with an infill matrix of lower density material and sintered. Different tungsten grades include different amounts of matrix material infilling between sintered tungsten particles. The properties and volume percentage of the infill material greatly affect the bulk material properties, with more matrix material generally reducing strength, reducing brittleness (increasing ductility), and reducing bulk density. Because the goal of this project was to have maximum density and maximum strength with a brittle fracture, a high purity tungsten material was chosen. The material is 99.95% pure tungsten rod from Goodfellow Corporation in the form of 10mm and 15mm diameter, centerless-ground rod.

Generally, tungsten is very difficult to machine because of its high hardness, especially in high purity versions. EDM is the preferred method of creating parts from tungsten because the process is largely independent of mechanical properties and instead depends primarily on electrical properties. μ WEDM also imparts very little force to the part during machining which is advantageous when creating small features that could be easily deformed. However, the process utilizes a stream of deionized water to flush the eroded material from the area being machined. In the initial testing, it was determined that this flushing pressure was causing parts to break off the substrate, so changes in flushing pressure and direction were implemented to produce a better part yield.

The modeled geometry which would exhibit the desired failure properties required that the substrate be cut into an array of blocks measuring $\sim 500\mu\text{m}$ on a side with each block being positioned on small blade of $100\mu\text{m}$ or smaller.

A challenge of the μWEDM process is that the machining geometry must be machinable using a continuous wire, i.e. there can be no blind features. Sometimes it is possible to create blind features by tilting the part so that the continuous wire passes through the part along a different axis. The block-on-blade geometry was achieved by cutting the parts in a 2-step process. First, the pillars and blocks were machined through the thickness of the part from one side, and then the part was rotated 90° , and the process was repeated on the second side as shown in FIGURE 2, thus producing an array of blocks on blades.

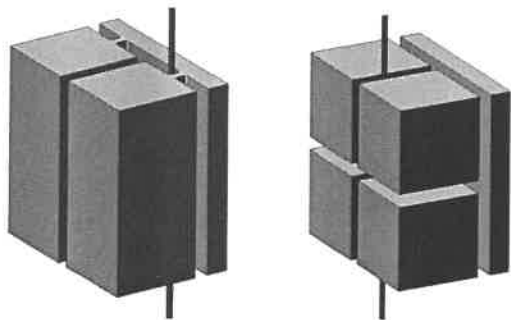


FIGURE 2. The wire EDM process uses a wire (shown in black) to cut half of the geometry (left). The part is then rotated 90° , and cut again (right) to create the final block-on-blade geometry.

One of the key design priorities for the project is to have the maximum possible mass of material included in the blocks. One factor in achieving this goal is to have the smallest possible gaps between blocks. The μWEDM machine used for this project is an Agie Vertex 1F which is capable of using $20\mu\text{m}$ diameter tungsten wire for cutting. A general rule of thumb for macro-scale EDM is that the minimum possible gap width that can be created by the process is 1.5 times the wire diameter. Initial testing, therefore, attempted to utilize $20\mu\text{m}$ wire to make the part with $30\mu\text{m}$ gap widths all around the blocks. This narrow gap creates a number of challenges for the machining process. First, the highest accuracy is achieved if the process employs a rough pass followed by one or more "trim" or

finishing passes. Because of the desire for a narrow gap, finishing passes were not used initially. Secondly, the machine creates the best surfaces and the most accurate geometry if it is only cutting on one side of the wire. This is only possible on a trim pass, so the initial parts did not benefit from this approach. Third, the flushing stream diameter is a constant diameter, so a narrower gap causes more of the flushing jet to impinge upon the block geometry which reduces flushing effectiveness and increases forces on the part. These initial process choices resulted in poorly formed blocks, widely varying blade dimensions and bent blades. Continued process development finally determined that a $76\mu\text{m}$ gap utilized multiple passes, produced highly accurate geometry and allowed adequate process flushing while preserving high part mass. Additionally, this large gap allowed the corners around the blade to be filleted with a radius of $25\mu\text{m}$. This fillet radius caused the process to create the fillet by moving the wire rather than by pausing and allowing the corner to be created by the wire geometry. When machining a full profile, the spark erosion of the μWEDM process occurs at the point at which the wire is closest to the geometry. But when the wire geometry is used to create a corner, all of the points around the wire are the same distance from the wire creating an indeterminate (and inaccurate) solution. The $76\mu\text{m}$ groove width also minimized the creation of "slugs" of unwanted material that fall into the water and create electrical shorts and wire breaks. This process development produced accurate parts as shown in FIGURE 3 with blade dimensions having less variation than the $\pm 1.5\mu\text{m}$ uncertainty of the optical measurement system.

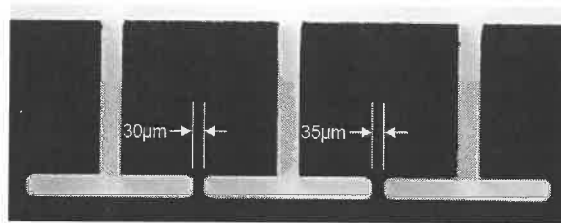


FIGURE 3. The final process parameters utilized $424\mu\text{m}$ wide cubes, $30\mu\text{m}$ wire, $76\mu\text{m}$ gap widths, filleted inside corners, and 1 main and 1 trim pass. The process was tested on different width blades.

MODEL VALIDATION AND TESTING

Once the $\mu\text{wire EDM}$ process parameters had been significantly improved, a set of tests was designed to validate the computational design

and analysis for blade sizes. The model required validation and tuning because of uncertainty in the material properties, especially across the very small blades where bulk material properties might not apply. In this test, parts were created with different size blades in the shear direction and the parts were impacted at specific shear loadings on a drop tower. The width of the blades increased row-by-row in 5 μ m increments from 25 μ m at the bottom of the part to 60 μ m at the top. All of the blade widths in the direction perpendicular to the shear direction were 60 μ m and each blade was 76 μ m tall (determined by the minimum machining gap). The purpose of the test was to shock the part at a particular loading and to have several rows of blocks with smaller blades shear from the part by failure of the blades while blades with larger dimensions would remain attached. Four parts were created with this geometry and tested on a drop table at different loadings. FIGURE 4 shows one of the parts after its loading on the drop table. Note that instead of whole rows shearing, there was some variability across the rows. It was expected that this variability would be due to dimensional variation from the manufacturing process.

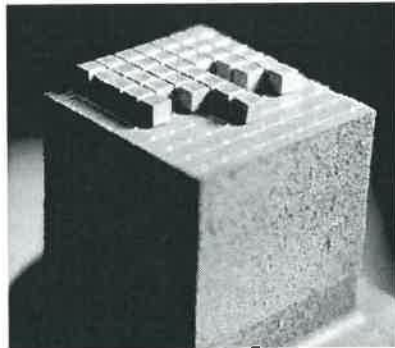


FIGURE 4. A part after shock testing on the drop table shows that several rows of smaller dimension blades sheared and rows with larger dimension blades remained intact.

It had proven very difficult to measure the blades after machining because of the blade being a minimum of 180 μ m down in a 76 μ m wide slot which obscured viewing. After testing, all of the blocks were picked off and the remaining broken blades were inspected using optical gaging. Additionally, SEM micrographs of the fracture surface were recorded for analysis of the fracture surface. An example SEM micrograph of the fracture surface is shown in FIGURE 5. The measurements were then analyzed to find the error in the average block from the design

width and to find any correlations between the blade width measurement's standard deviation and the design width or column number. In all of the measurements, it was estimated that the measurement uncertainty was 1.5 μ m due to difficulty in picking the measurement points. As can be seen in FIGURE 5, the cut lines at the edges of the blade are difficult to determine in this post-failure state. Therefore, any errors or standard deviations under this value are not significant. The analysis found that the average values for manufactured blades were very close to the design values and comparable to the measurement uncertainty. The standard deviation was generally on the order of the uncertainty, and plots of the data did not indicate any statistically significant trend connecting deviation to either blade width (for the variable width direction) or in column number (for the fixed 60 μ m direction).

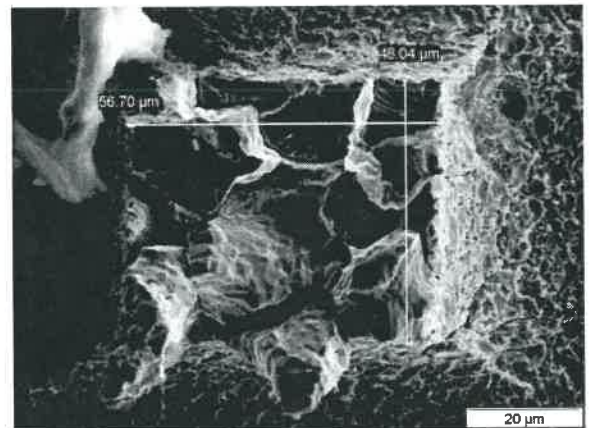


FIGURE 5. Fracture surface from a blade feature shows the cracks in the substrate material (background) and the cracks between tungsten particles in the blades.

While the measurements showed the blade feature geometry to have been manufactured with better than anticipated accuracy, the SEM study did show the fracture surfaces to contain unexpected features. The micrographs showed cracks across the part surface which seem to correlate with powder particle boundaries in the original tungsten material. These cracks are most likely due to the powder metallurgy processes used to create tungsten material. Because the tungsten used is 99.95% pure, there is very little matrix material between the particles. The particles are sintered together, but the sintered contact area on each particle appears to be much smaller than the total surface area of the particle. Many of the fracture

zones appear to have whole particles broken out of the material rather than typical fracture patterns as would be expected of a more homogeneous material. Inspection of the sintered surfaces areas showed shear bands indicative of brittle fracture. It was concluded that this particle to particle fracture was the cause for most of the variability in shear loading response of the blades.

FINAL DESIGN

These results were used to statistically determine the optimum blade dimensions to be $45\mu\text{m} \times 45\mu\text{m}$ and to determine shock loadings which would result in over 99% certainty that the blades failure state would be predictable. Because of the variability in the test results (due to the material), the shock results had a wider spread than was desired with 300G loading for no blade failure and 3000G loading for blade failure in excess of 99%. Even with the large spread, the concept would still be demonstrated, so a set of parts was created with these design parameters and the parts were tested. FIGURE 6 shows a single image extracted from the high speed video of one parts undergoing high G loading. The part is clamped in a fixture and this image, taken as the drop table impacted the stops, shows that all of the blocks have separated from the tungsten part and are falling away from the part.

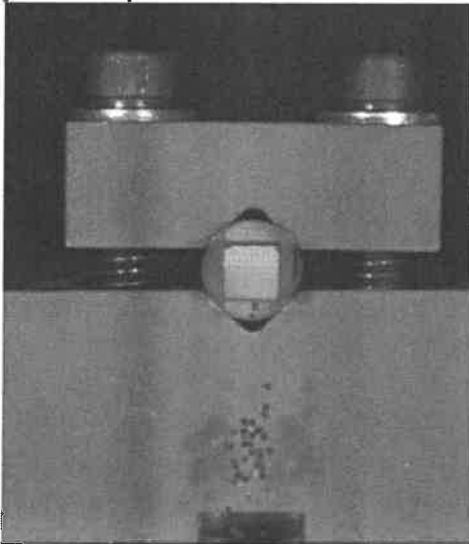


FIGURE 6. Image extracted from the high speed video of an engineered material part undergoing high-G testing. The blocks have separated from the part upon impact and are shown falling from the part.

This testing showed that none of the blocks failed below 300G loading as predicted. However, during the 3000G testing, it was discovered that the shear failure of the parts was dependent on the loading duration. Parts loaded for 0.5ms did not always fail, but parts loaded for 1.5ms did always fail. When a part is dependent on the load duration, this is indicative of an energy dependence for failure which points to ductile failure. So, while the material failure surfaces indicated microscopic, localized brittle failure, the separation of the particles was causing the bulk material property to be slightly ductile instead of brittle. Despite this somewhat ductile response, the ability to create an engineered material was demonstrated.

CONCLUSIONS

The results demonstrated the ability to create an engineered material by using μWEDM to form a precise geometry in the part surface. The material and part geometry met the desired criteria for predictable shear failure, high density, and high repeatability. The μWEDM process produced excellent repeatability and accuracy of feature dimensions and did so without distorting the delicate features during the manufacturing process. The particles from the powder metallurgy process used to make the tungsten are believed to have caused the bulk material failure to be somewhat ductile, though evidence of localized brittle failure was observed in the fracture surfaces. To further improve the performance of the engineered material, a more brittle failure would need to be achieved, either by utilizing non-powder metallurgy tungsten, or by using a different material or a composite of different materials.

ACKNOWLEDGEMENTS

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FABRICATION OF A NEW THREE DIMENSIONAL POSITION SURVEYING SYSTEM FOR THE UNDERGROUND LONG DISTANCE PIPES USING FOUR ROTARY ENCODERS

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INTRODUCTION

Optical fiber cable for broadband communication is usually covered by plastic pipe and buried under the ground. Then, we are required to grasp its real buried position before digging at the construction or the maintenance in order to prevent from the cutting of the optical fiber. However, once the optical fiber cable is buried under the ground, it is very difficult to confirm the buried position of the pipe, because the optical fiber cable and plastic cover pipe are not the metal.

A former way to confirm the buried position of the optical fiber cable is the ground penetrating radar method.^[1] The ground penetrating radar method is very useful for the metal. However, the optical fiber cable in plastic cover pipe is not metal. Another method is the elastic wave method.^[2] In this method, a vibrator is put on the ground. The wave from the vibrator is detected by many in-lined receivers which are arrayed on the ground. However, the detection is only done at the place where the ground is covered by soil and the accuracy of the surveying is not good at the place where the position of the plastic cover pipe is so deep in the ground or in the different soils.

If we can detect directly the information of the buried three dimensional position of the pipe by a new position surveying system that can move itself in the pipe, the problem of the former way to confirm the buried position of the optical fiber cable is solved. This research aims to identify

easily the buried position of the pipe, using the information of the buried position from the pipe. We use four rotary encoders that have pulleys contacting top, bottom, right and left walls of the buried pipe in order to detect directly the information of the buried position of the pipe.

It is confirmed that the new position surveying system can identify the buried position of a pipe that is 20 m long and has loose three-dimensional curvature in the error of 39 mm

PRINCIPLE OF POSITION SURVEYING

Length of walls of both side of a pipe is same if the pipe is straight. If the pipe curves toward right, the length of the right wall is shorter than the length of the left wall. We use four rotary encoders that have pulleys contacting top, bottom, right and left walls of the buried pipe in order to detect directly the information of the buried position of the pipe. Output of rotary encoder contacting the wall shows the length of the wall. Average of summation of the outputs of the top, the bottom, the right and the left rotary encoder shows moving distance of a surveying mechanism of the position surveying system. The curvature of vertical direction of the buried pipe is obtained by difference of the outputs of the top and the bottom rotary encoder. The curvature of horizontal direction of the buried pipe is obtained by difference of the outputs of the right and the left rotary encoder. The three dimensional position of buried pipe is identified by the moving distance and the curvature of horizontal direction and the curvature of vertical

direction. The principle of position surveying of horizontal direction using the right and the left rotary encoders is shown in Fig. 1. Inner diameter of the pipe that has loose curvature is D . The original point is P_0 . Present position of the surveying mechanism is P_n . Angle between the tangential line and the x-axis is θ_z . It is assumed that a difference ε is made by the both side rotary encoders when the surveying mechanism moves a small distance of s . This moved point is P_{n+1} . The normal lines that cross points P_n and P_{n+1} make an angle θ_z . The radius of the curvature is assumed as R . The difference ε of the right and left rotary encoders is shown as

$$\varepsilon_z = \left(s + \frac{\varepsilon_z}{2}\right) - \left(s - \frac{\varepsilon_z}{2}\right) = \left(R_z + \frac{D}{2}\right)\theta_z - \left(R_z - \frac{D}{2}\right)\theta_z = D\theta_z \quad (1)$$

Then, the small angle θ_z is obtained as

$$\theta_z = \frac{\varepsilon_z}{D} \quad (2)$$

Similarly, the curvature of vertical direction is obtained by difference of the outputs of the top and the bottom rotary encoder. The small angle of vertical direction is θ_y . The surveying mechanism may incline by some rolling force. The rolling angle is assumed as θ_x . The $P_{n+1}(x_{n+1}, y_{n+1}, z_{n+1})$ are calculated from the $P_n(x_n, y_n, z_n)$, θ_x , θ_y , θ_z and s .

The coordinates of $P_{n+1}(x', y', z')$ after z rotation is

$$x'_{n+1} = s \cdot \cos(\theta_z + \theta_z) \quad (3)$$

$$y'_{n+1} = s \cdot \sin(\theta_z + \theta_z) \quad (4)$$

$$z'_{n+1} = 0 \quad (5)$$

The coordinates of $P_{n+1}(x'', y'', z'')$ after y rotation is

$$x''_{n+1} = x'_{n+1} \cdot \cos(\theta_y + \theta_y) \quad (6)$$

$$y''_{n+1} = y'_{n+1} \quad (7)$$

$$z''_{n+1} = x'_{n+1} \cdot \sin(\theta_y + \theta_y) \quad (8)$$

The coordinates of $P_{n+1}(x, y, z)$ is what is added the coordinates after X rotation to the past coordinate of the point $P_n(x, y, z)$

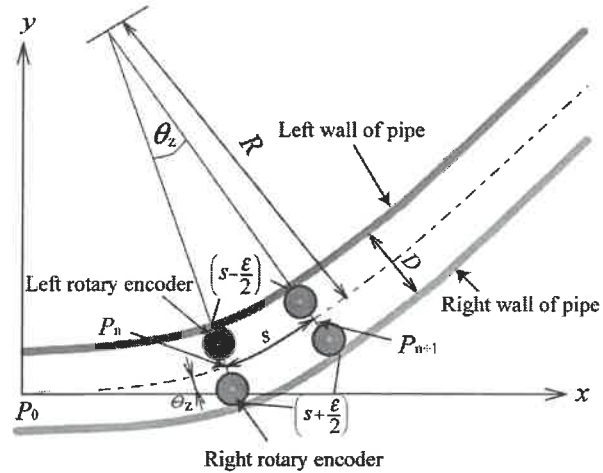


FIGURE 1. The principle of the position surveying in the underground pipe by in-pipe surveyor using rotary encoders

$$x_{n+1} = x_n + x''_{n+1} \quad (9)$$

$$y_{n+1} = y_n + y''_{n+1} \cdot \cos(\theta_x + \theta_x) - z''_{n+1} \cdot \sin(\theta_x + \theta_x) \quad (10)$$

$$z_{n+1} = z_n + y''_{n+1} \cdot \sin(\theta_x + \theta_x) + z''_{n+1} \cdot \cos(\theta_x + \theta_x) \quad (11)$$

The tangential angle Θ is the summation of the small angle θ and shown as

$$\Theta_x = \sum \theta_x \quad (12)$$

$$\Theta_y = \sum \theta_y \quad (13)$$

$$\Theta_z = \sum \theta_z \quad (14)$$

POSITION SURVEYING SYSTEM

A new position surveying system for underground pipes is shown in Fig. 2. The system consists of a surveying mechanism that carries the four rotary encoders, a signal processor, an I/O interface and a computer. The surveying mechanism that is 375 mm long is held to the inside of the pipe by eight small rollers and can move in the pipe. However, it is difficult to measure the length of the bottom inside pipe, because the cable exists in the pipe. Consequently, the rotary encoder is inclined 45 degrees and attached to the surveying mechanism. Small angle θ_{x0} is 45 degrees. A weight is attached under chassis in order to prevent rolling of the surveying mechanism. However, rolling of the mechanism can not be prevented perfectly. Therefore, rolling angle θ_x is

measured by another rotary encoder that is attached the chassis. The rotary encoders are held by arms that have pin joints and are attached the chassis of the surveying mechanism. The pulley is pushed to the wall of the pipe by force of 1 N and vibration of the rotary encoder is severely damped, because the arm is suspended by a spring and a dashpot. The circumference of a pulley is 64.7 mm. The signal processor consists of a detecting circuit of clockwise or counter clockwise and a mechanical chattering circuit. The rotary encoder makes 1440 pulses per one rotation. Resolution of the rotary encoder is 44 μ m. The signal from the signal processor is introduced into the computer through the I/O interface. Software in the computer calculates locus of the surveying mechanism every 64.7 mm. This method decreases an error by the distortion of the pulley. Consequently, Small distance s is 64.7 mm of the same length as the circumference of a pulley. The result of the position surveying is displayed on the screen of the computer.

PIPE FOR SURVEYING EXPERIMENT

Pipe for surveying experiment is shown in Fig. 3. The pipe consists of four straight pipes and two circular arc pipes. The diameter of the pipe is 44 mm. The first and second straight pipe is 4 m long. The third circular arc pipe has a horizontal curvature radius of 30 m and is 4 m long. The fourth and sixth straight pipe is 2 m long. The fifth circular arc pipe has a vertical curvature radius of 20 m and is 4 m long. Pipe for experiment measured coordinates on the basis of the laser level. The x-coordinates at 20 m of the y-axis is 1702 mm and z-axis is 284.5 mm.

RESULT OF POSITION SURVEYING

We measure the three dimensional position coordinates of the pipe for surveying experiment by the surveying system. An error is included

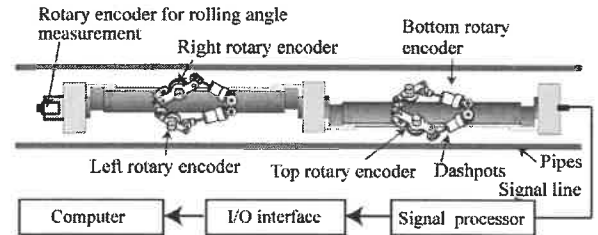


FIGURE 2. Structure of the position surveying system

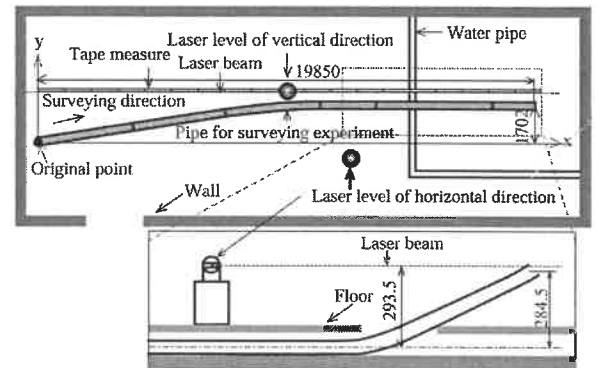


FIGURE 3. Pipe for surveying experiment

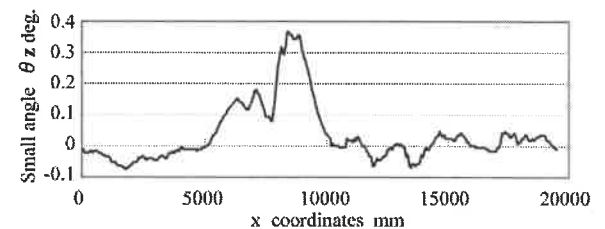


FIGURE 4. The small angle θ_z after using dynamic threshold processing and averaging processing

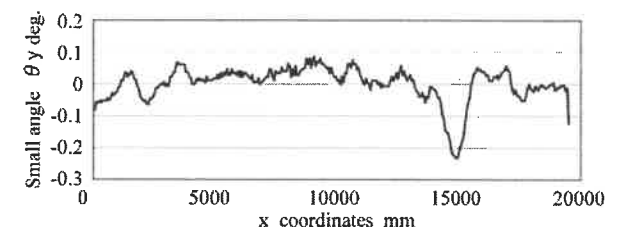


FIGURE 5. The small angle θ_y after using dynamic threshold processing and averaging processing

in the small angle calculated from the difference in pair rotary encoder pulse. Therefore, the error included in the small angle is eliminated using the average processing and the dynamic threshold processing. The small angle θ_z eliminated the error is shown in Fig. 4. In the horizontal circular arc pipe of the x-coordinates at 9000 mm, the small angle exists 0.35 degrees. The small angle θ_y that removed the error is shown in Fig. 5. In the vertical circular arc pipe of the x-coordinates at 15000 mm, the small angle exists 0.22 degrees. The position coordinates of horizontal direction by the position surveying system and the position coordinate of horizontal direction by the steel measure are shown in Fig. 6. The position coordinates of vertical direction by the position surveying system and the position coordinate of vertical direction by the steel measure are shown in Fig. 7. The vertical maximum of difference occurs at the curved pipe and is 39 mm. The horizontal maximum of difference occurs at 14m of x-coordinates and is 30.6 mm. The error of vertical direction by the position surveying system without using the dynamic threshold processing and the average processing is 50 mm. The error of horizontal direction by the position surveying system without using the dynamic threshold processing and the average processing is 250 mm. Consequently, it was confirmed that the dynamic threshold processing and the averaging processing is very effective.

CONCLUSIONS

1. We fabricated of a new three dimensional position surveying system for the underground pipes using four rotary encoders. Four of rotary encoders measure distance of top, bottom, right and left walls of the pipe.

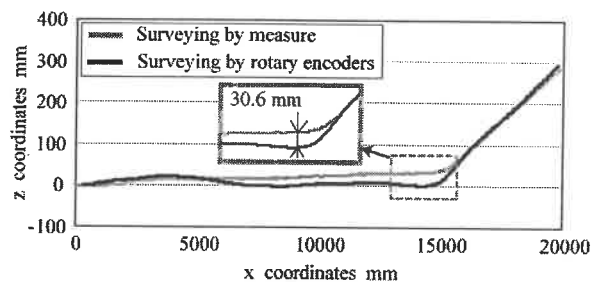


FIGURE 6. Measurement result of horizontal direction

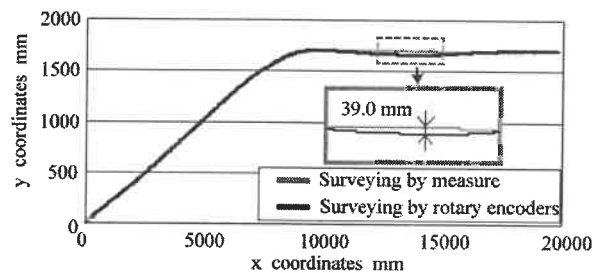


FIGURE 7. Measurement result of vertical direction

2. The curvature of vertical direction of the buried pipe is obtained by difference of the outputs of the top and the bottom rotary encoder. The curvature of horizontal direction of the buried pipe is obtained by of the outputs of the right and the left rotary encoder.
3. It is confirmed that the new position surveying system can identify the buried position of a pipe that is 20 m long and has loose three-dimensional curvature in the error of 39 mm.

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AN ANALYTICAL MODEL FOR ORBITAL DRILLING AND ITS APPLICABILITY

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1. INTRODUCTION

Recently, composite materials are widely used as structural materials in aerospace and aircraft industries because of their remarkable advantages such as lightweight and high strength. However, those materials are mostly difficult to machine in conventional ways. Therefore, more effective and practical machining method is required to solve the problems. The orbital drilling method is one of solution.

Machining mechanism of the orbital drilling is similar to helical milling by machining center (FIGURE.1). Orbital drilling equipment has two axes, which are eccentrically settled each other. A cutting tool is rotated on tool spindle, settled eccentrically from the orbital spindle. Tool revolution spindle and orbital spindle is independent. Helical drilling motion is performed by XY linear actuator. On the other hand, the orbital drilling is performed by mechanical circular motion device is rotating symmetry. Therefore, holes machined by the orbital drilling

have higher roundness than holes machined by helical drilling. The orbital drilling can drill in composite materials effectively and fast. Besides thrust loading on orbital drilling is low in comparison with conventional way. For example we can drill holes in CFRP without surface exfoliation. Furthermore, surface machined by the orbital drilling is smooth enough and requires no finishing process.

Today, the orbital drilling technology has been adopted in specified industrial field. Besides, the influence of milling condition on holes geometry, the fact of efficient drilling and lower thrust loading are not considered. It's also important to analyze principle for the orbital drilling technology developing.

The objective of this study is considering the factors which make thrust loading lower on drilling.

2. CUTTING TOOL MOTION

In order to identify control factors on orbital drilling, clarifying cutting tool motion is necessary. We focused bottom edge motion on cutting tool which inferentially affects thrust loading and construct geometric cutting model of orbital drilling.

Cutting edge on bottom face is expressed as 4 position vectors simulation, as show in FIGURE 2.

Vector r_o stands for position from orbital center to tool center and its value depend on orbital equipment control.

Vector r_s stands for position from tool center to cutting edge on bottom face. Vector ϵ_o and ϵ_s are unexpected eccentricity and include repeatable

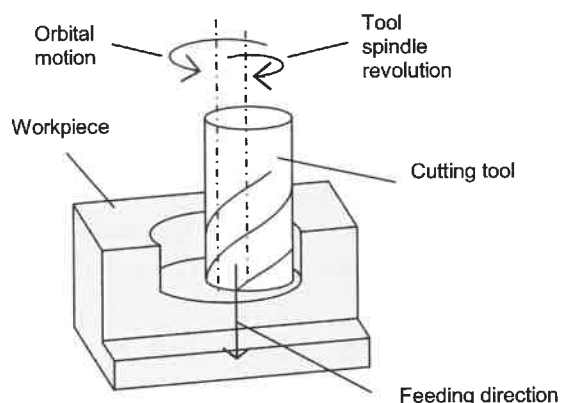


FIGURE 1. Cutting tool motion in orbital drilling/helical milling

unit and non repeatable run out. ϵ_o is caused by orbital center and ϵ_s is by tool center. In this study ϵ_o and ϵ_s are ignored because we consider ideal cutting tool motion without run out.

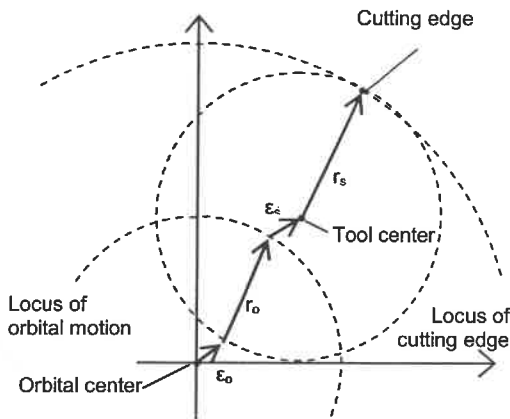


FIGURE 2. Geometrical model for orbital drilling

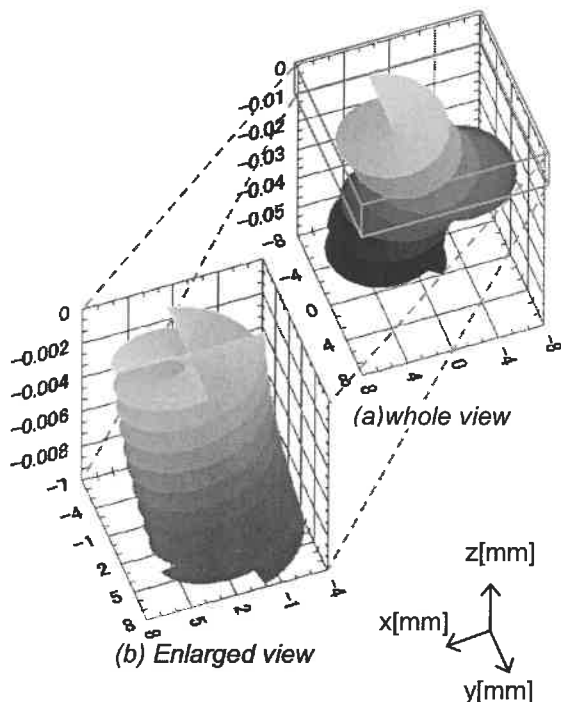


FIGURE 3. Locus of cutting edge motion on the bottom face

TABLE 1. Simulation parameters

Orbital speed	300 min ⁻¹
Tool spindle speed	24700 min ⁻¹
Orbital eccentricity	4 mm
Tool diameter	10 mm
Feeding speed to axial direction	100 mm/min

2.1 Simulation of bottom edge motion

We analyze bottom edge motion with computer analyzing program developed on LabVIEW (National instruments). TABLE 1 shows parameters of orbital drilling on the simulation. They are almost same parameters which are set on actual orbital drilling into CFRP. The intended cutting tool is square end-mill.

Result of the simulation is illustrated in FIGURE 3. FIGURE 3(a) shows only a cutting edge motion and enlarged by z-axis (tool axial direction). FIGURE 3(b) shows all 4 cutting edge motion and more enlarged. They show locus of accumulation of cutting edge motion, which make a helical shape. We guess that volume of the helical shape is volume of eliminated work piece by bottom cutting edges and the shape affect thrust loading.

2.2 Bottom cutting edge velocity

Velocity of a cutting edge on bottom face depends on orbital speed (tool speed around orbital center) and spindle speed (tool revolution speed) (FIGURE 4 (a)). Therefore the velocity is not constant from speed and direction. An angle at the line to tool center to orbital center and a line to tool center to the point is named angler difference is the significant factor. Direction of tool revolution around orbital center is CCW and direction of tool revolution speed on its center axis is CW.

Velocity of the cutting edge by tool revolution (v_s) is defined by the eq. (1) where, r_s stands for tool radius and N_s stands for spindle speed.

$$v_s = 2\pi N_s r_s \quad (1)$$

Velocity of the point by orbital speed (v_o) is defined by the eq. (2) where, r_o stands for orbital eccentricity and N_o stands for orbital speed/

$$v_o = 2\pi N_o r_o \quad (2)$$

From e.q. (1) and (2), velocity of cutting edge is calculated and the result is showed in FIGURE 4. TABLE 2 shows calculation parameter. In the case of these parameters, N_o and r_o is larger than actual orbital drilling parameter because of clarifying effect of orbital motion. FIGURE 2(b) shows the result by angler difference considered and FIGURE 2(c) shows the result by no angler difference. Because v_s and v_o cancel them each

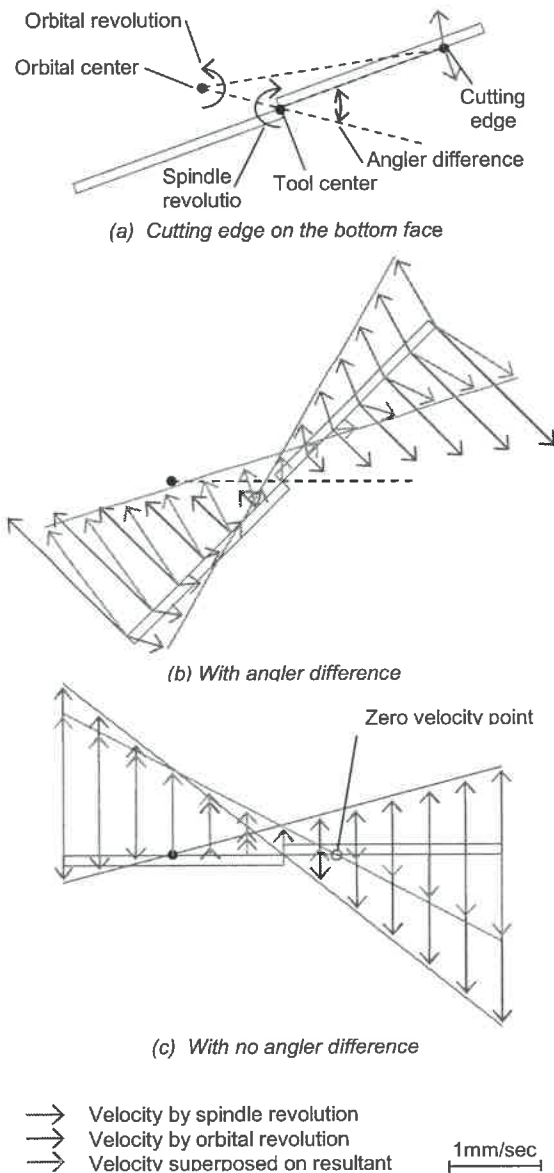


FIGURE 4. Velocity of the point on bottom edge other. When it is no angle difference, zero velocity point on bottom face. It makes thrust loading larger.

t stands for where, time from angle difference becomes 0 to next 0 angle difference. The relationship between t and angle difference is

$$2\pi N_s t + 2\pi = 2\pi N_o t \quad (3)$$

Where,

$$t = 1 / (N_s - N_o) \quad (4)$$

Therefore t is inverse related to the difference between N_s and N_o . However it means feeding speed to axial direction is too fast from spindle speed, so thrust load can't obtain small. It is

essential to avoid zero velocity point. The actually special cutting tools for orbital drilling have no cutting edge around tool center.

3. IMPULSE PER UNIT VOLUME

In order to consider orbital drilling and thrust load while drilling, helical milling experiment was carried out as shown in FIGURE 5. Large thrust load while drilling means inefficient drilling in terms of machining energy loss and causes flaked surface on composite material.

Thrust load accumulation during drilling time e.g. impulse per unit volume (I/UV) is considered to analyzing the drilling efficiency the impulse could be related to drilling energy loss.

3.1 Thrust load measurement

Thrust load while helical drilling hole (diameter: 15mm, depth: 20mm) into aluminium alloy is measured by using of diameter. TABLE 3 shows instruments for the experiment. We use the experimental conditions (TABLE 4) on the basis of orthogonal array L_9 concatenating.

FIGURE.6 shows an example of measured thrust load, FIGURE 7 shows the appearance of helical drilled holes and TABLE 4 shows result of Impulse per unit volume.

Table 2. velocity analyzing parameters

Tool diameter	12.0 mm
Orbital eccentricity	3.0 mm
Tool spindle speed	3000 min ⁻¹
Orbital speed	1000 min ⁻¹

Table 3. Mechanical tool and equipment

Machining center	Matuura FX-1
Tool dynamometer	Kistler 9256A
Charge amp	Kistler 9256C1
Cutting tool	Mitsubishi, coated carbide endmill (2 flute)



FIGURE 5. Machining experiment view

TABLE 4. Orthogonal array for experiments

Exam No.	OS [min ⁻¹]	TSS [min ⁻¹]	FS [mm/min]	TD [mm]	impulse / unit volume [N·sec/m ³]
1	30	1000	20	8	58345020.4
2	30	5000	30	10	48491750.1
3	30	10000	40	12	11179254.7
4	60	1000	30	12	14028686.2
5	60	5000	40	8	19453585.3
6	60	10000	20	10	22432438.5
7	90	1000	40	10	28268644.2
8	90	5000	20	12	22815606.0
9	90	10000	30	8	20451658.9

OS: Orbital Speed, TSS: Tool Spindle Speed
FS: Feed Speed, TD: Tool Diameter

TABLE 5. Auxiliary condition

factor	Level 1	Level 2	Level 3
OS	OS1	OS2	OS3
TSS	TSS1	TSS2	TSS3
FS	FS1	FS2	FS3
TD	TD1	TD2	TD3

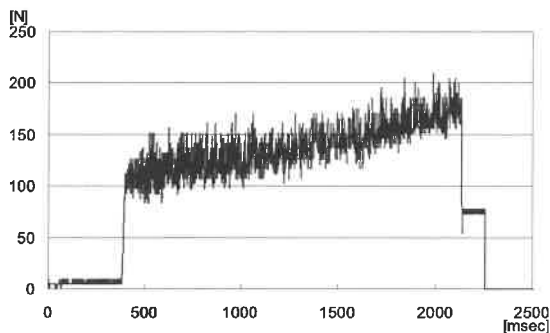
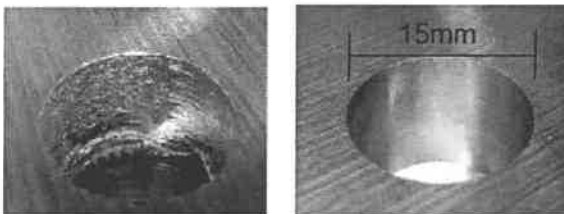


FIGURE 6. Thrust load in helical milling



(a) Rough

(b) Smooth

FIGURE 7. Hole surface of machined surface

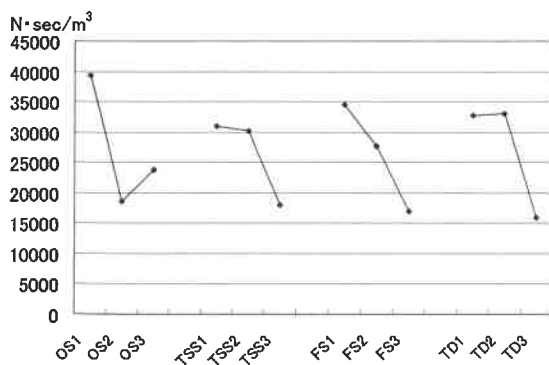


FIGURE 8. Functional effect diagram

3.2 Consideration

From the experimental results, Auxiliary condition (TABLE 5) and Factorial effect diagram (FIGURE 8). The auxiliary conditions one calculated, OS2, TSS3, FS3 or TD3 make I/UV small and OS1, TSS1 (TSS2), FS1 or TD2 (TD1) make I/UV large.

In the case of ex.No.1, parameters, which make I/UV large (OS1, TSS1, FS1 and TD1) is the largest I/UV made rough hole surface (FIGURE 7(a)). As the result of 3 parameters, which I/UV small (TSS3, FS3 and TD3) the smallest I/UV and made smooth hole surface (FIGURE 7(b)). Therefore small I/UV makes smooth hole surface.

However we guess trends between parameters and I/UV from FIGURE 7 is not always correct. E.g. ratio of OS to TSS to FS and ratio of TD to hole diameter could also affect to the fine drilling.

4. Conclusion

In this study, we have developed a geometrical cutting model, simulated bottom edge motion, analyzing bottom edge velocity and measured thrust load. We obtained the following concluding remarks.

1. Bottom edge motion making spiral shape
2. Bottom edge having zero speed point and better shape of no edge around tool center
3. Impulse per unit volume affecting hole surface roughness.

On a future work, we will develop simulation program for analyzing bottom edge eliminates work piece and experimental orbital drilling.

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REDUNDANCY METHOD FOR THE SEPARATION OF MACHINE AND TRIGGERING PROBE ERRORS IN SMALL VOLUME MEASUREMENT ON A CMM

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ABSTRACT

Coordinate measuring machines (CMMs) are subject to periodic verification to ensure measurement capability. However, users also need to rapidly diagnose the source of accuracy degradation for corrective action. This paper presents a means to separate errors coming from the triggering probe from those due to the rest of the machine using multiple redundancy probing of the machine's own test sphere. The method effectiveness is demonstrated for a variety of stylus length and shows its sensitivity to changes of lobbing errors.

INTRODUCTION

Coordinate measuring machines are widely used in industry to control dimensions and geometry on mechanical parts. Several sources of error affect the accuracy of CMMs such as errors from the axes of the machine and errors from the probing system [1]. These errors coexist. When measuring a gauge block, a ring gauge or a master ball in a small volume to evaluate the probing system, it is usually assumed that the effect of the axes of the machine are negligible so that the observed errors are essentially caused by the probing system [2]. The bias that characterizes the probe system is the pre-travel variation. Pre-travel variation is the change in the effective spatial delay between the machine position when contact occurs and the machine position read by the machine at trigger, the point of measurement [3, 4]. To assess these errors without any machine influence, Mayer and al. [5] used a micrometer to contact the probe tip and a video camera and macro lens to visualize the initial contact and so measure the pre-travel in 2D. Dobosz and Wozniak [6] developed a setup which moves the stylus tip through a low force system. The initial contact just before the stylus tip moves is detected by a high sensitivity mechanical system which then continues moving the stylus tip until triggering occurs. Pre-travel variation can be a dominant error source when using a CMM to inspect small features [7].

It has also been established that the probe errors generally increase with the length of the stylus [3, 8]. Despite these developments there remains an industrial requirement to assess probe performance in a production environment and on the CMM and determine whether the probe or the rest of the machine are causing erroneous readings. Redundancy methods are often used in the metrology field to separate error contributions from a stable but unknown artifact and from a measuring system [9]. The work presented in this article describes a method to separate the errors of the probing system from those due to the rest of the machine by measuring a test sphere through a multiple redundancy approach.

MEASUREMENT OF A SPHERE AND RESIDUAL ANALYSIS

Let us assume n measurements are taken on a test sphere using a touch trigger probe on a CMM. Once the effect of sphere location is removed through least squares fitting a sphere to the data, each radial residual e is indicative of three main error groups. These error groups are the geometric form error of the sphere δ_{fs} , the machine axes motion errors and out-of-squarednesses δ_{am} and the probing system pre-travel variation δ_p .

These radial errors are assumed here to algebraically cumulate into the total error as follow :

$$e = \delta_{fs} + \delta_{am} - \delta_p. \quad (1)$$

Figure 1 shows the probing system and the rest of the machine consisting of the machines axes (and their associated electronics, drives and encoders) and the test sphere.

With the probe in its vertical orientation it becomes possible to fully probe the upper hemisphere of the test sphere. The probe can then be rotated nominally around its axis using the B angle of the indexing head and another series of points acquired on the test sphere. Since only the probe (and indexing head) has rotated between the two series of

measurements it is assumed that the error contributing elements that did not rotate such as the test sphere sphericity and the machine axes will remain the same. We amalgamate these two error sources and refer to them as the errors originating from the machine, δ_m , so that we can write $\delta_m = \delta_{fs} + \delta_{am}$.

Equation (1) then becomes,

$$e = \delta_m - \delta_p \quad (2)$$

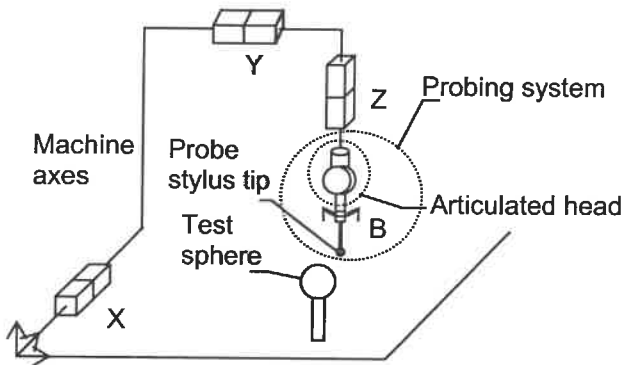


Figure 1 : Probing system and test sphere

REDUNDANCY METHOD

The target contact points on the sphere are equally spaced on a pre-defined number of parallel circles between the equator and the pole. n points are measured for each circle equally spread longitudinally. The sphere is measured for n configurations of the probing system, each configuration defined by rotation angle B around the vertical axis (Renishaw PH10M articulation system) at increments of $360^\circ/n$. For every configuration, the probe system remains vertical. From one configuration to another, a probe qualification is first performed on the sphere. As a result of the rotation there is a permutation of the probe errors on all the points measured. However, the machine errors are assumed to remain the same. Figure 2 illustrates the measurement sequence for one circle parallel to the equator of the test sphere for the different configurations.

For example for configuration 1, the machine will probe the sphere nominally normally along n direction j , $j = 1, \dots, n$ corresponding to n touch probe direction i , $i = 1, \dots, n$. So a direction set (j, i) is used to describe each touch. For configuration 2, the probe has been rotated by one increment, so that this time while the machine still moves along directions j , $j = 1, \dots, n$ the probe direction are now i with $i = n, 1, 2, \dots, n-1$. This process continues until all n rotations have been used.

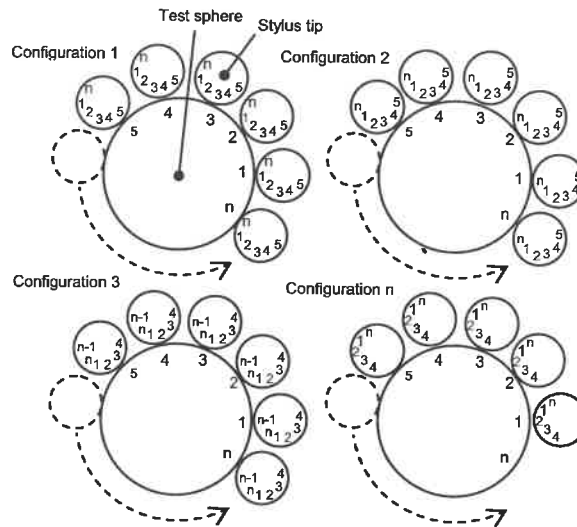


Figure 2 : Illustration of the multiple redundancy method used

IDENTIFICATION MATRIX

For one configuration of the probing system, there are n measured points on the circle, thus combining n machine errors and n probe errors. For each measured point, the following equation applies, $e_{j,i} = \delta_{m,j} - \delta_{p,i}$ (3) where j : indicates the machine approach direction and i : indicates the touch probe direction.

For each configuration and a particular circle, we have a sub-system of n equations. When considering n configurations, a linear system results from the concatenation of all configuration sub-systems.

The system rank is deficient by 1. To solve the system and have the absolute probe errors and the machine errors, one more equation would be needed. However, for a probing system what matters is the variation of the probing error. So, we proceed arbitrarily to set the n^{th} machine error to zero by removing the last column from the system as well as the associated error term. We then have an identification matrix of full rank with a good condition number of about 10. The system can now be solved for the remaining errors in the system, keeping in mind that these values are floating, with an offset or bias which depends on the error which was set to zero.

EXPERIMENTAL TESTS

Two circles are used, the first circle is the equator of the sphere, the second circle is at a latitude of 30° . For each configuration, 24 points ($n=24$) are used for each circle, Figure 3.

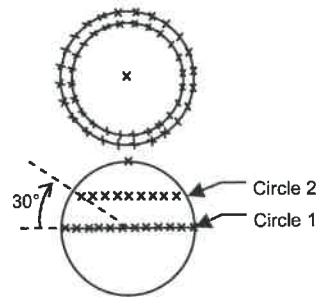


Figure 3 : Measured points on the sphere test

Tests are performed on a moving-bridge CMM model LK G90c. It uses a TP2 trigger probe mounted on an articulated system model PH10M. The test sphere has a diameter of 20 mm. The sequence is repeated at four positions within the machine measurement volume to assess if this has an influence on the machine error results. Figure 4 shows the four test sphere positions. For position 1, the test is further repeated using six different stylus lengths: 20, 40, 60, 80, 110 and 130 mm in order to analyze the impact on the probe error results.

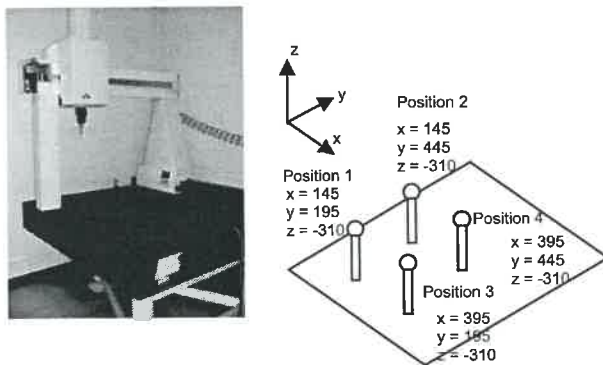


Figure 4 : a) Photo of the CMM model LKG90c;
b) the four positions of the sphere in the machine reference frame.

Figure 5 presents the estimated machine and probe error contributions for the test sphere in position 1 and the three stylus lengths used (20, 60 and 110 mm). The results for the probe errors are in good agreement with the known fact that probe lobing error is strongly dependent on stylus bending itself strongly influenced by the stylus length [3, 8]. The results for the machine errors are more surprising since it was expected that this error would be much smaller than the probe errors. They exhibit an oval shape with amplitude comparable to the probe error with the shorter stylus. However, as expected, the estimated machine error remains

virtually the same regardless of the length of the stylus. For measuring positions 2, 3 and 4, the sphere was measured only with a stylus length of 20 mm. The estimated errors remained almost the same, both for the probe and the machine.

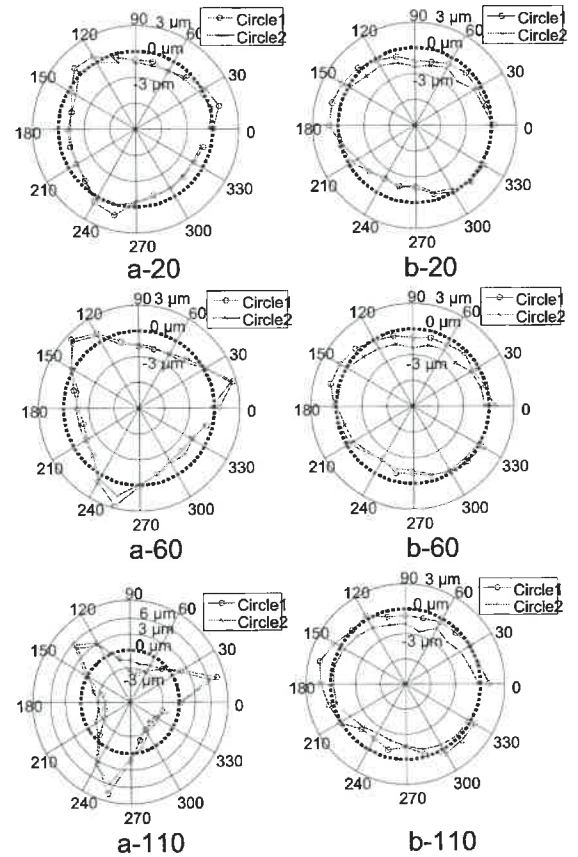


Figure 5 : Measurement results of the test sphere in position 1 with the 3 lengths stylus (20, 60 and 110mm)

CONCLUSION

A practical method to separate triggering probe error (rotated during the tests) from the error of the rest of the machine (not rotated) has been proposed. The method is based on redundancy measurement of a test sphere. The method was tested in the laboratory. The advantage of the method is that it can be easily performed on a commercial device in industrial setting. The processing software could be integrated in the vast majority of CMMs. It allows automatic mode to be used and no extra hardware, simply the reference sphere generally present on a CMM table. By separating probe and machine errors it facilitates remedial actions should excessive errors be detected.

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FABRICATION OF Pt/C MULTILAYER-COATED THIN FOIL MIRRORS FOR HARD X-RAY TELESCOPE LOADED IN ASTRO-H SATELLITE

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INTRODUCTION

In future satellite missions for X-ray astronomy, hard X-ray imaging will become one of the major observation techniques. It allows astronomers to observe the distribution of hard X-ray objects in the Universe with a high sensitivity. Since hard X-rays can penetrate thick absorbing materials, astronomers are able to discover many massive black holes embedded in thick gas. High-energy phenomena are well studied in hard X-rays, which are essentially produced by accelerated high-energy particles in the universe. The sixth Japanese X-ray astronomy satellite ASTRO-H [1] to be launched in 2013 is expected to provide the first opportunity to perform hard X-ray imaging observations with a supermirror hard X-ray telescope [2]. Similar hard X-ray imaging systems have also been proposed for use in other programs, such as the Italian-French mission Symbol-X [3] (planned for 2014), the European Space Agency (ESA)-Japan X-ray observatory mission XEUS [4] and the future NASA mission Constellation-X [5]. They are now merged into the International X-ray Observatory IXO to be launched in 2020. A hard X-ray telescope is included in the list of instruments to be onboard IXO.

Next-generation hard X-ray telescopes require highly nested aspherical thin mirrors of 100 nm shape accuracy and less than 0.3 nm rms surface roughness [1, 6]. These telescopes are constructed using nested multiple grazing incidence reflecting mirrors consisting of a paraboloid and hyperboloid of revolution, as shown in Figure 1. Over two hundred Pt/C multilayer-coated mirrors of different sizes are required [1]. Thin mirror substrates or shells are fabricated by replication from molding dies of high precision. Thus, a machining apparatus that enables the accurate and rapid machining of

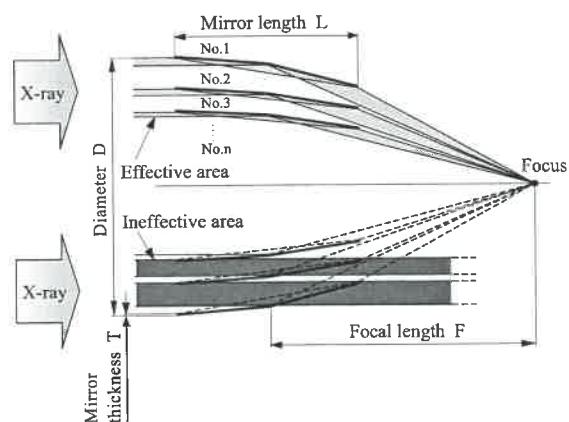


FIGURE 1. Structure of a grazing incidence X-ray telescope [4].

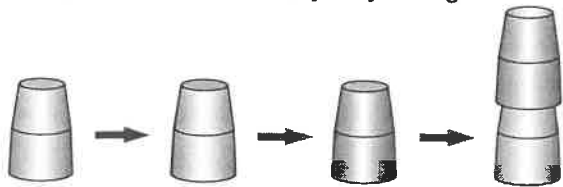
large aspherical molding dies is necessary, and the shapes of such large molding dies should be measured on the machining apparatus during machining to ensure shape accuracy.

In this paper, we describe the present status of fabrication processes for making Pt/C multilayer-coated thin foil mirrors for next generation hard X-ray telescope loaded in ASTRO-H satellite.

IDEAL MANUFACTURING PROCESS

Figure 1 shows the Wolter type-1 optics for X-ray telescope, which has paraboloid and hyperboloid nested thin mirrors. Multiple pairs of aspherical mirrors have to be nested coaxially and confocally. It is essential that the thickness of mirror substrates is thin for obtaining high aperture efficiency and light weight. And the inner side of aspheric mirrors is coated with Pt/C multilayer supermirror [1, 2]. Figure 2 shows the manufacturing process of hard X-ray aspherical thin foil mirrors. In this case, we need more than

two hundred aspherical molding dies of different sizes in high precision. If we use glass molding dies in high precision, the machining cost is very high and we will meet with a problem of separation between molding dies and substrates, though the mirror accuracy may be high.



Molding die Pt/C coating Covering Separation
substrate

FIGURE 2. Manufacturing process of X-ray mirrors.

PRESENT MANUFACTURING PROCESS

We are now using glass tubes in the market as molding dies instead of aspherical molding dies in high precision. The tube was washed and coated with Pt/C multilayer supermirrors by RF ion beam sputtering as shown in Figure 3. Aluminum thin foils of 0.2 mm thick were bent by mechanical rollers, glued on the coated glass tube, and cured in a heated vacuum chamber. Then the thin foil having Pt/C multilayer was separated from the glass tube as shown in Figure 3. This method is very cheap and simple, though we can get the very smooth Pt/C multilayer surface on aluminum substrates. We do not need any intermediate layer between glass tube and multilayer for separation. The mirror shape is conical approximation. The weak point of this mirror is lack of shape accuracy, though this mirror is lightest in the world.



Glass tube Pt/C coating Covering Separation
Al substrate

FIGURE 3. Manufacturing process of X-ray mirrors from glass tubes.

Figure 4 shows a real hard X-ray telescope which was launched in the space by a balloon. The mirror is one-thirds in circumference, and there are two stages in the telescope, so that X-ray will reflect by double conical mirrors.

NEXT GENERATION MANUFACTURING PROCESS

Figure 5 shows the future manufacturing process. In this case the molding die is made of

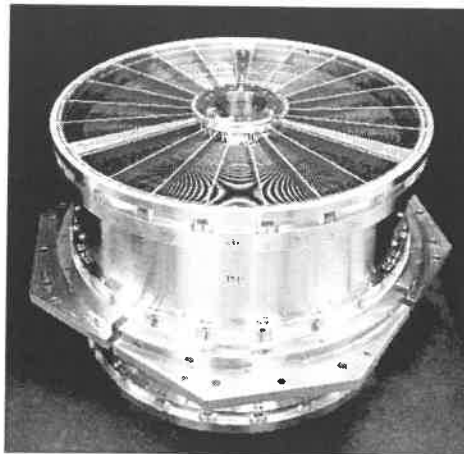


FIGURE 4. Real hard X-ray telescope launched by a balloon.

electroless nickel coated on aluminum alloy, and diamond turned for making shape, then polished for obtaining super-smooth surface. There are two kinds of shapes, that is, paraboloid and hyperboloid in various diameters. This process has a feature of cost and time cut for making the super precise aspherical molding die as well as much better figure accuracy compared with present process. The biggest problem will be separation of Pt/C multilayer from electroless nickel molding dies.



Molding die Pt/C coating Covering Separation
Al substrate

FIGURE 5. Manufacturing process of X-ray mirrors.

Machining of Aspherical Molding Dies

Figure 6 shows the large ultra-precision diamond turning, polishing and measuring machine newly developed for fabricating aspherical electroless nickel molding dies of 400 mm mirror height and 600 mm diameter. The machine is supported by three air stages. It has a large rotary table (C stage) of 830 mm diameter supported by a precise air bearing controlled with 0.001 degree resolution, and horizontal (Z) and vertical (Y) axes driven by linear motors in the range of 420 mm ultra-precision V-V roller slides with 1 nm resolution. Two high-precision linear laser scales with 0.14 nm resolution are fitted on the Y- and Z-axes. The temperatures of cooling oil and clean air are controlled within $296.2 \text{ K} \pm 0.1 \text{ K}$ to maintain the

stability and repeatability of the large ultra-precision machine. The machine is controlled by a CNC system. The machine was designed for watertight ultra-precision polishing and washing processes after SPDT. A polishing head can be set on the tool post.

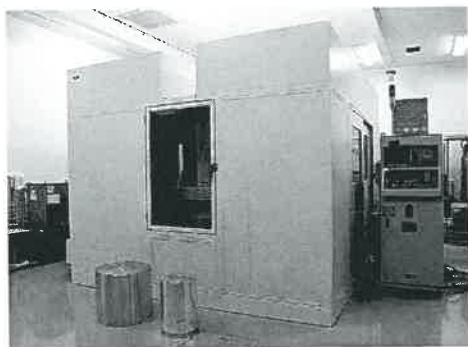


FIGURE 6. Newly developed large ultra-precision diamond turning machine of vertical spindle on a vibration proof foundation in a clean room.

Next-generation hard X-ray telescope mirrors consist of grazing incidence reflecting thin mirrors with a paraboloid and hyperboloid section, as shown in Figure 1. It is necessary to machine an aspherical surface on an aluminum alloy molding die with a mirror length of 400 mm and a diameter of 300 mm to prepare such mirrors. Figure 7 shows a shape accuracy of ± 392 nm after machining a molding die with a paraboloid and hyperboloid section without any compensation. Using this data for compensation, we recut the same sample on the machine and obtained ± 92 nm shape accuracy, as shown in Figure 7.

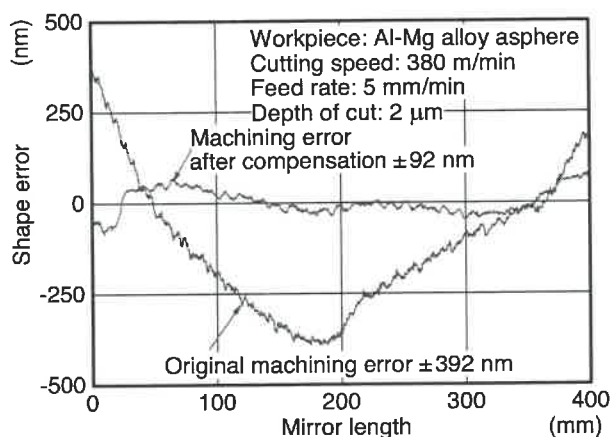


FIGURE 7. Shape accuracy of double aspherical surfaces of aluminum alloy die.

Figure 8 shows three electroless nickel molding dies of 300 mm diameter and 210 mm mirror length. The upper part of 10 mm in length is hyperboloid and the lower part of 200 mm in length is paraboloid. The picture shows an electroless nickel coated aluminum alloy die, a diamond turned die and a polished die from the left to right. The surface roughness of last one is 0.3 nm rms.

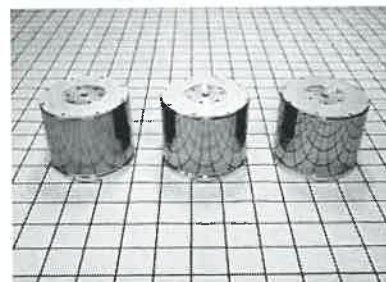


FIGURE 8. Electroless nickel plated aluminum die, diamond turned and polished molding dies.

Mirror Manufacturing

Figure 9 shows a photograph of Pt/C multilayer supermirror coated aspherical molding die just opening the RF ion beam sputtering system in Nagoya University.

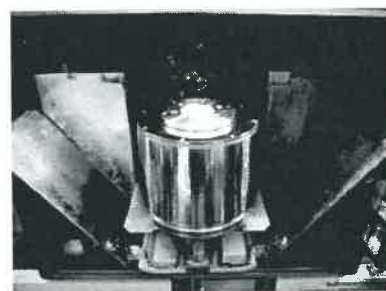


FIGURE 9. Photograph of Pt/C coated aspherical molding die after opening the vacuum chamber.

Aluminum thin foil was glued on the Pt/C coated aspherical molding die as shown in Figure 9, then cured in a heated vacuum chamber with load. Figure 10 shows aluminum foil on the Pt/C coated aspherical molding die after curing. Figure 11 shows two kinds of Pt/C multilayer mirrors after separation. These aspherical mirrors are made by replication technique as shown in Figure 5. For separation the intermediate layer between the Pt/C multilayer supermirror and electroless nickel molding die is essential.



FIGURE 10. Photograph of aluminum thin foil substrate on a Pt/C coated aspherical molding die after curing.

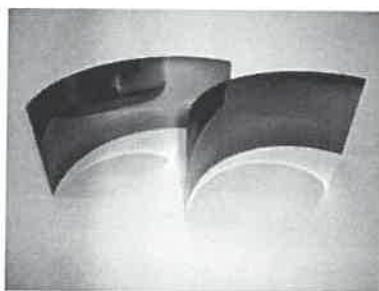


FIGURE 11. Photograph of Pt/C coated aspherical X-ray mirrors.

Figure 12 shows photographs of single-point diamond-turned aspherical molding dies for the ASTRO-H application (300 mm diameter, 12 m focal length) and the future IXO application (500 mm diameter, 60 m focal length). Those aluminum molding dies of 400 mm mirror length have to be coated with electroless nickel for obtaining smoother surfaces.

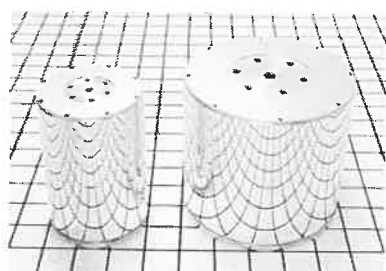


FIGURE 12. Photograph of diamond turned double aspherical dies of 400 mm mirror length, and 300 mm diameter on the left and 500 mm diameter on the right.

SUMMARY

Next-generation hard X-ray telescopes require highly nested aspherical thin mirrors of 100 nm shape accuracy and less than 0.3 nm rms surface roughness. These telescopes are

constructed using nested multiple grazing incidence reflecting mirrors consisting of a paraboloid and hyperboloid of revolution. Over two hundred Pt/C multilayer-coated mirrors of different sizes are required. Thin mirror substrates or shells are fabricated by replication from molding dies of high precision. Thus, a machining apparatus that enables the accurate and rapid machining of large aspherical molding dies is necessary. We describe a large ultra-precision single-point diamond turning, polishing and measuring machine for fabricating aspherical molding dies of high precision, which can be applied to next-generation hard X-ray telescope mirrors. We also describe the Pt/C coated thin foil mirrors replicated from the aspherical molding dies.

ACKNOWLEDGEMENTS

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THE LATEST APPLICATION TECHNOLOGIES IN MICRO EDM

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INTRODUCTION

Micro mechanical components and tooling are required to support advances in many fields including electronics, medical and optics industries. Makino has long been a leader in manufacturing high precision machine tools. We are putting our accumulated knowledge of many years to the test with several new machines that are designed to achieve ultimate accuracy in both Wire EDM and Ram EDM. We would like to introduce some of Makino's advanced technologies, which have been proven in the field.

Ultra Precision Wire EDM "UPN-01"

Wire EDM for micro mechanical components requires the use of a very fine wire to achieve the minimum kerf and smallest corner radius. The Makino UPN-01 is unique having the wire travel horizontally through a submerged oil bath. This machine is capable of automatically threading fine wire of 20 μ m (0.0008 in) through a 30 μ m (0.0012 in) start hole with close pitch dimension to adjacent holes. The minimum kerf possible is 16.9 μ m using a 15 μ m (0.0006 in) tungsten wire.

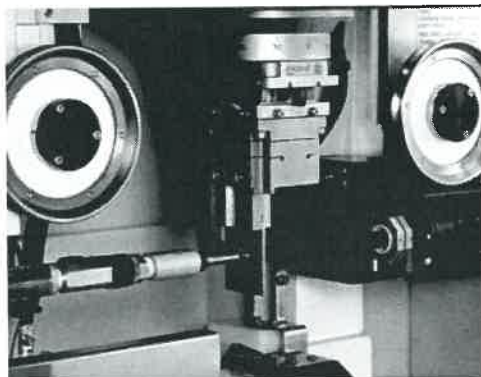


Photo 1. An automatic threading nozzle which carries a wire electrode and a suction nozzle are brought close to both sides of the work-piece.

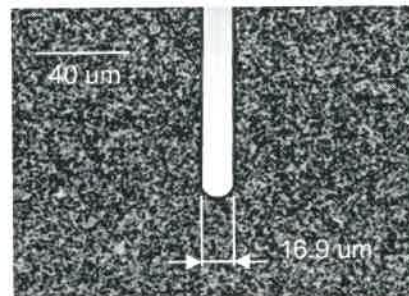


Photo 2. Minimum kerf with 15 μ m tungsten wire. Carbide 1.5mm, Surface finish 0.5 μ mRz, 1 cut

Minimum feed increment of 0.1 μ m or less is achieved by aerostatic pressure guide-ways.

The machine is also designed to use no lubrication to achieve minimum impact on the environment.

Constant temperature technology for controlling the ambient air temperature is used to minimize thermal drift during cutting. Tightly sealed machine covers and a high-performance air conditioner control the temperature variation in different parts of the machine to within ± 0.5 C. Maintaining the internal-machine temperature constant also helps to reduce the cost of air conditioning the entire shop.

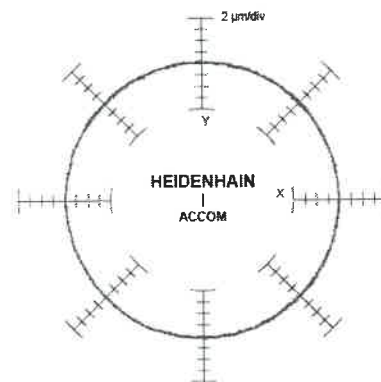
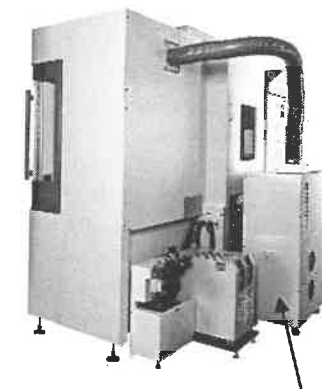
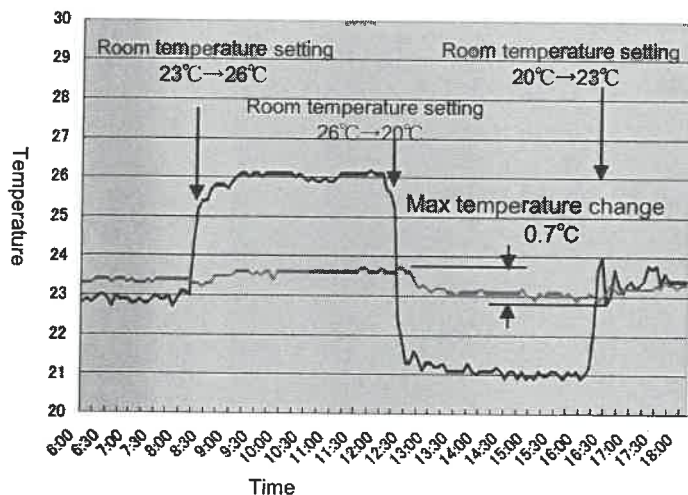


Figure 1. Static accuracy for XY axes Roundness 0.6 μ m, Diameter 5mm



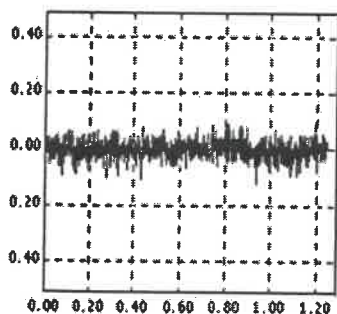
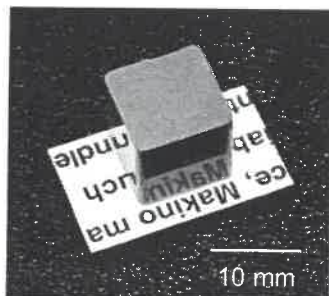
Air conditioner for thermal chamber

Figure 2. Internal machine temperature is always controlled within $\pm 0.5^\circ\text{C}$ even if the room temperature changes widely.

SPG machining circuit (Makino-designed machining circuit which provides micro pulses under the certain condition) provides the best surface finish of $0.14\mu\text{mRz}$ in carbide materials. Brass wire which has a copper / zinc ratio in the range of 65/35 to 63/37 is by far the most widely used wire type and is also used for the sample in Fig3. Incidentally we usually recommend using brass wire due to cost performance.

All begins with through temperature control

The UPN-01 has been assembled in a special facility called UP-Room. The UP-Room has no cranes that could cause vibration and is divided into several booths where temperature of each machine can be rigorously controlled. Separated booths make it possible to assemble and inspect individual machine under temperature conditions identical to the customer's shop floors.



Ra 0.021um
Ry 0.172um
Rz 0.140um

Figure 3. Best surface finish with SPG machining circuit. Carbide 10mm, Wire type 0.1mm brass. Measured by KOSAKA SURFCORDER SE3500.



Photo 3. UP-Room, UPN-01 and EDAC1

Ultra Precision RAM EDM "EDAC1"

As digital cameras and mobile phones become more compact and lightweight, the size of electronic components and connectors that are incorporated into these devices, also must be reduced. This requires smaller corner radius for die/mold tooling of these products. RAM EDM is utilized for blind features and precision small holes where Wire EDM cannot reach. Developed for these applications is the EDAC1.

The EDAC1 is able to achieve minimum corner radius of 5 μ m in steel. The best surface finish capability is 0.45 μ mRz in carbide using a copper tungsten electrode and an interface level difference within $\pm 1\mu$ m in the X and Y plane. Z height differences of $\pm 1\mu$ m are achieved between several EDMed surfaces. These high accuracies are achieved using an advanced Z-axis stabilizer. In high-accuracy machining, even the slightest amount of heat generation cannot be tolerated. The Z-axis ball screw is constantly cooled by oil, which reduces the sharp rise in temperature caused by repeated jump motions. For all axes, full travel perpendicularity and straightness are held to within $\pm 2\mu$ m.

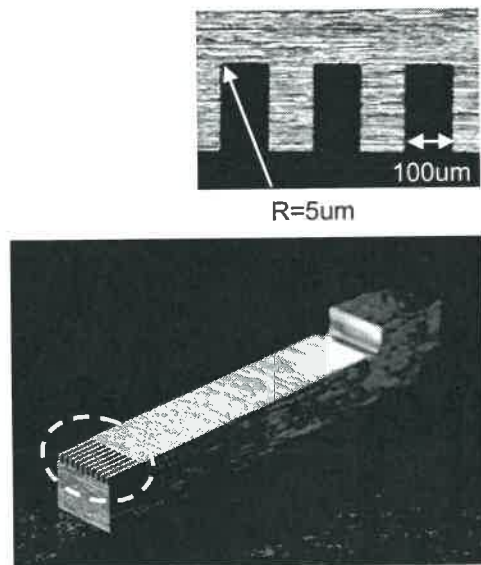


Photo 4. Ability to execute tiny corner radius is in increasing demand for precision die/mold, electronic connectors and other micro components.

Work-piece: Steel (SKD11, JIS)
Electrodes made of copper tungsten are made by UPN-01
Surface finish: 0.6 μ mRz with SPG circuit

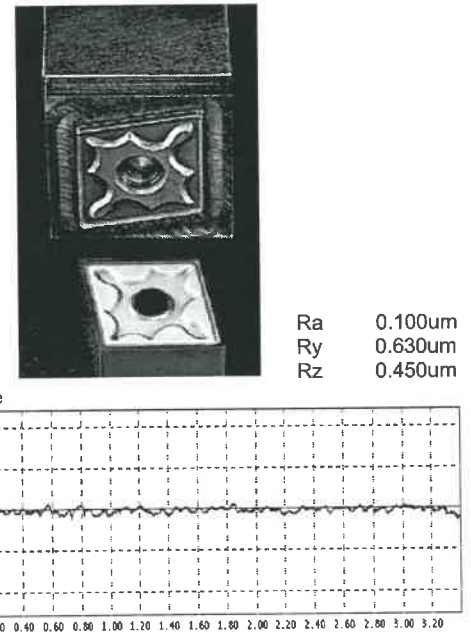


Figure 4. Best surface finish in carbide.
Electrodes made of copper tungsten are produced by Makino's vertical Machining Center V33i.
Surface finish: 0.45 μ mRz with SPG circuit.
Measured by KOSAKA SURFCORDER SE3500.

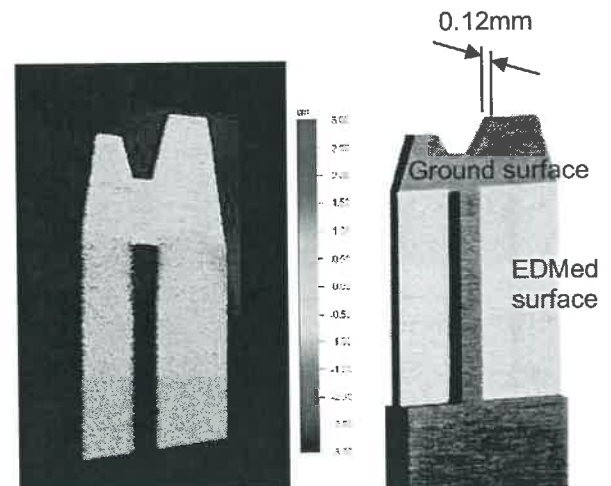


Figure 5. Interface level difference between ground and EDMed surface improves in $\pm 1\mu$ m.

Work-piece: Steel (SKD11, JIS)
Electrode: Copper
Surface finish: 0.8 μ mRz with SPG circuit
Measured by non-contact 3D measuring device, WYKO NT series NT8000

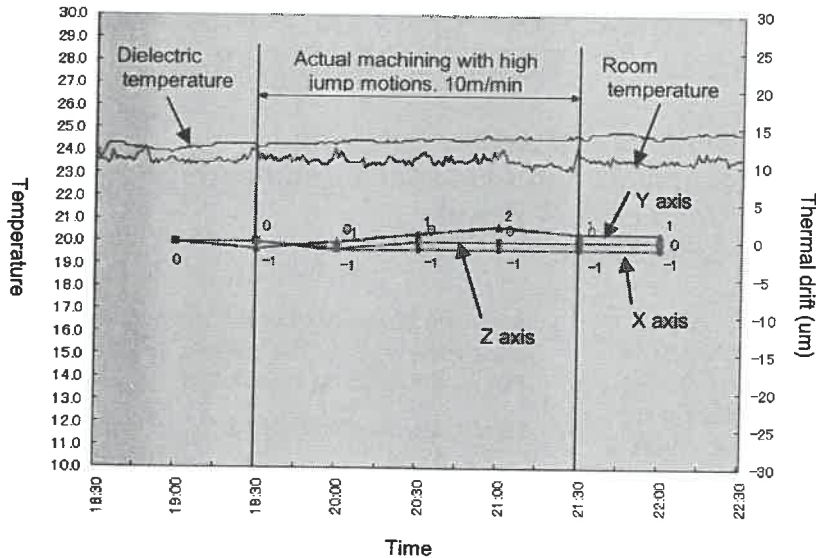


Figure 6. The Z-axis stabilizer reliably removes heat produced by repeated high jump motions.

Fine Hole EDMing

In order to produce fine holes precisely, it is very important to improve rotational deviation of spindle head. The MA head which is limited the rotational deviation within 2 μ m makes perfect positioning possible.

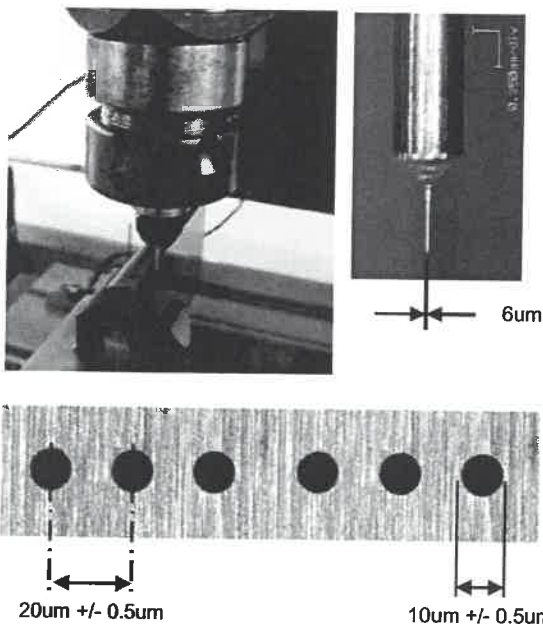


Photo 5. Finest holes with dressed electrodes of 6 μ m.

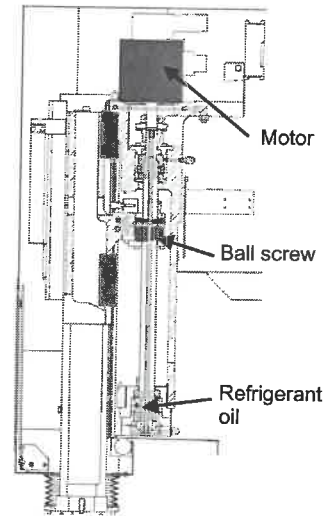


Figure 7. Heat sources of Z-axis (Motor, Ball screw). These are constantly cooled by refrigerant oil.

It is possible to choose a suitable solution from 2 kinds of available spindle heads. One is an MA head which is mainly used for fine hole machining with dressed or hollow pipe electrodes, and the other is an Mi head fitted with a rotary encoder to guarantee indexing accuracy of ± 2 sec. Both heads help to automate the micromachining process.

CONCLUSION

Companies all over the world are facing financial crisis and the influence of the expanding economic recession. In response to demand for new innovations from our customers Makino is committed to providing new ideas for micro EDM. It is our aim to foster innovation that will lead to engineering of new high value products that previously could not be made without our technology.

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APPLICATION OF RECEPTANCE COUPLING TO TORSIONAL AND AXIAL FREQUENCY RESPONSE PREDICTION

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INTRODUCTION

Many research studies have identified the relationship between the dynamics of the cutting tool-machine-workpiece and the machining process. Due to the cutting force, time-dependent deflections between the tool and workpiece are obtained. The relationship between the resulting deflection and applied force is described by the system frequency response function, or receptance. Depending on the force direction and structure geometry, the relevant receptances may describe (lateral) bending, torsion, and/or axial vibration behavior.

While experimental techniques may often be applied to determine the required receptances directly, in some instances this approach may prove inconvenient. In these situations, a preferred alternative is to implement a predictive technique based on tool-machine models. Challenges to this paradigm include damping estimation, contact stiffness determination, and full geometric knowledge of commercial components. To address these obstacles, the Receptance Coupling Substructure Analysis (RCSA) procedure was developed [1] by building on the principles detailed by Bishop and Johnson [2]. Using RCSA, models of the tool and holder are coupled to a measurement of the spindle-machine to predict the assembly's bending receptances. This information may then be used, together with milling models, to predict the stability and forced vibration behavior.

In this paper, the RCSA method is extended to prediction of torsional and axial receptances. This is particularly important for drilling, where torsional-axial vibration coupling can lead to chip thickness variation and chatter. Comparisons between experimental and predicted results for torsional and axial receptances of a stepped diameter free-free beam are provided.

TORSIONAL RECEPTANCES

The torsional receptances, $s_{ij}(\omega)$, for a cylindrical, constant cross-section beam with

free-free boundary conditions may be expressed as shown in Eqs. 1-4, where ϕ_i is rotation about the beam axis at coordinate i , t_j is an external torque at coordinate j , $\lambda = \omega \sqrt{\frac{\rho}{G}}$, ρ is density, G is shear modulus, l is beam length, and J is the second polar moment of area [2]. See Fig. 1.

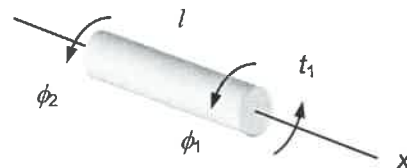


FIGURE 1. Circular cross-section beam with torque applied at coordinate ϕ_1 .

$$s_{11}(\omega) = \frac{\phi_1}{t_1} = \frac{-\cot(\lambda l)}{GJ\lambda} \quad (1)$$

$$s_{21}(\omega) = \frac{\phi_2}{t_1} = \frac{-\csc(\lambda l)}{GJ\lambda} \quad (2)$$

$$s_{12}(\omega) = \frac{\phi_1}{t_2} = \frac{-\csc(\lambda l)}{GJ\lambda} \quad (3)$$

$$s_{22}(\omega) = \frac{\phi_2}{t_2} = \frac{-\cot(\lambda l)}{GJ\lambda} \quad (4)$$

As depicted in Fig. 2, once the individual receptances are known, two components I and II may be coupled to produce the assembly III. If the direct receptance at assembly coordinate 1 is desired, an external harmonic torque with magnitude T_1 is applied at assembly coordinate 1, labeled Φ_1 (note the use of lower case variables for components and upper case variables for assemblies). In Fig. 2, it is assumed that coordinates ϕ_1 (on component I) and Φ_1 (on assembly III) are spatially collocated before and after the rigid coupling. The coupling coordinates are ϕ_{2a} and ϕ_{2b} . The free-free component receptances are: $s_{11} = \frac{\phi_1}{t_1}$, $s_{21} = \frac{\phi_2}{t_1}$,

$$s_{12} = \frac{\phi_1}{t_2}, \text{ and } s_{2a2a} = \frac{\phi_{2a}}{t_{2a}} \text{ for I; and } s_{2b2b} = \frac{\phi_{2b}}{t_{2b}},$$

$$s_{32b} = \frac{\phi_3}{t_{2b}}, s_{2b3} = \frac{\phi_{2b}}{t_3}, \text{ and } s_{33} = \frac{\phi_3}{t_3} \text{ for II. They may}$$

be calculated from Eqs. 1-4 by appropriate substitution. The compatibility condition for the rigid coupling is $\phi_{2b} - \phi_{2a} = 0$, which enables the equality $\phi_{2a} = \phi_{2b} = \Phi_2$ to be written (Φ_2 is assumed to be collocated with ϕ_{2a} before and after the rigid coupling). Also, $\phi_1 = \Phi_1$ and $\phi_3 = \Phi_3$. The equilibrium conditions are $t_{2a} + t_{2b} = 0$ and $t_1 = T_1$.

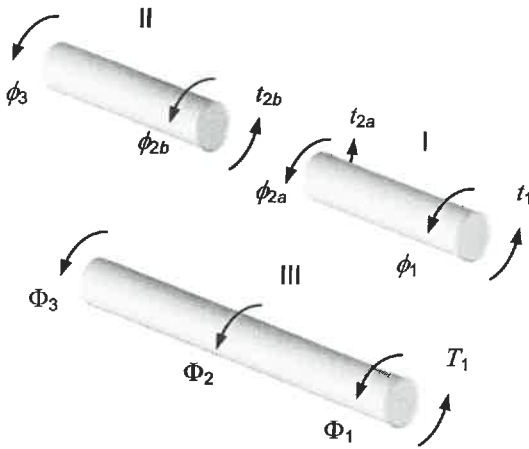


FIGURE 2. Rigid coupling of components I and II to form assembly III. An external torque T_1 is applied to determine the assembly receptances S_{11} and S_{31} .

To determine $S_{11} = \frac{\Phi_1}{T_1}$, the component torsional

rotations are first written. For I, there are two torques acting on the body, so the torsional rotations are:

$$\phi_1 = s_{11}t_1 + s_{12a}t_{2a} \text{ and } \phi_{2a} = s_{2a1}t_1 + s_{2a2a}t_{2a}. \quad (5)$$

For II, the torsional rotations are $\phi_{2b} = s_{2b2b}t_{2b}$ and $\phi_3 = s_{32b}t_{2b}$. Next, substitution into the compatibility condition gives:

$$s_{2b2b}t_{2b} - s_{2a1}t_1 - s_{2a2a}t_{2a} = 0. \quad (6)$$

Then, application of the equilibrium conditions enable t_1 to be replaced by T_1 and t_{2a} ($t_{2a} = -t_{2b}$) to be eliminated from Eq. 6. See Eq. 7.

$$s_{2b2b}t_{2b} - s_{2a1}T_1 + s_{2a2a}t_{2b} = 0 \quad (7)$$

Grouping terms in Eq. 7 enables t_{2b} to be written as $t_{2b} = (s_{2a2a} + s_{2b2b})^{-1} s_{2a1}T_1$. Correspondingly, $t_{2a} = -(s_{2a2a} + s_{2b2b})^{-1} s_{2a1}T_1$. Finally, substitution of this torque value into the S_{11} expression gives the desired result; see Eq. 8. Note that the assembly response is written as a function of the component direct (s_{11} , s_{2a2a} , and s_{2b2b}) and cross (s_{12a} and s_{2a1}) receptances. The expression for t_{2b} is used to determine the cross receptance, S_{31} . See Eq. 9.

$$S_{11} = \frac{\Phi_1}{T_1} = s_{11} - s_{12a}(s_{2a2a} + s_{2b2b})^{-1} s_{2a1} \quad (8)$$

$$S_{31} = \frac{\Phi_3}{T_1} = s_{32b}(s_{2a2a} + s_{2b2b})^{-1} s_{2a1} \quad (9)$$

The direct and cross receptances, S_{33} and S_{13} , respectively, may be determined by applying a torque at Φ_3 . See Eqs. 10 and 11.

$$S_{33} = \frac{\Phi_3}{T_3} = s_{33} - s_{32b}(s_{2a2a} + s_{2b2b})^{-1} s_{2b3} \quad (10)$$

$$S_{13} = \frac{\Phi_1}{T_3} = s_{12a}(s_{2a2a} + s_{2b2b})^{-1} s_{2b3} \quad (11)$$

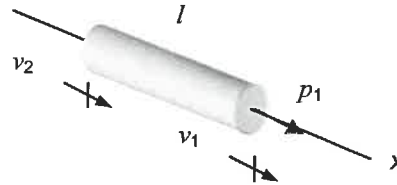


FIGURE 3. Beam with axial force p_1 applied at coordinate v_1 .

AXIAL RECEPTANCES

The axial receptances, $a_{ij}(\omega)$, for a constant cross-section beam with free-free boundary conditions may be expressed as shown in Eqs. 12-13, where v_i is the deflection along the beam axis at coordinate i , p_j is an axial force at coordinate j , $\lambda = \omega \sqrt{\frac{\rho}{E}}$, E is elastic modulus, and A is the cross-sectional area [2]. See Fig. 3.

$$a_{11}(\omega) = \frac{v_1}{p_1} = \frac{-\cot(\lambda l)}{EA\lambda} = a_{22}(\omega) = \frac{v_2}{p_2} \quad (12)$$

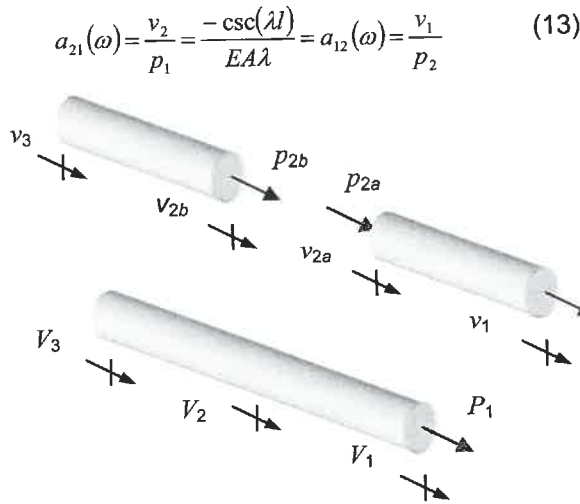


FIGURE 4. Rigid coupling of components I and II to form assembly III. An external axial force P_1 is applied at V_1 to determine the assembly receptances A_{11} and A_{31} .

Figure 4 depicts a rigid coupling example similar to the one shown in Fig. 2. The two free-free components I and II are coupled to produce the assembly III. To determine the direct receptance at the right end of the assembly, an external harmonic axial force with magnitude P_1 is applied at assembly coordinate 1, labeled V_1 . Using the same assumptions and following the same steps enables the assembly axial receptances to be written as a function of the component axial receptances. See Eqs. 14-17.

$$A_{11} = \frac{V_1}{P_1} = a_{11} - a_{12a}(a_{2a2a} + a_{2b2b})^{-1}a_{2a1} \quad (14)$$

$$A_{31} = \frac{V_3}{P_1} = a_{32b}(a_{2a2a} + a_{2b2b})^{-1}a_{2a1} \quad (15)$$

$$A_{33} = \frac{V_3}{P_3} = a_{33} - a_{32b}(a_{2a2a} + a_{2b2b})^{-1}a_{2b3} \quad (16)$$

$$A_{13} = \frac{V_1}{P_3} = a_{12a}(a_{2a2a} + a_{2b2b})^{-1}a_{2b3} \quad (17)$$

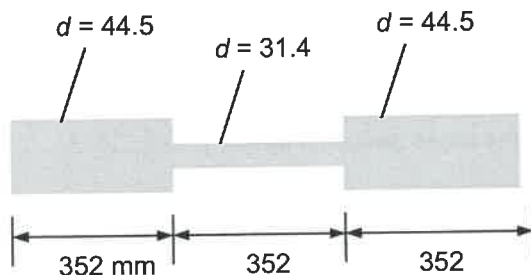


FIGURE 5: Stepped diameter beam geometry.

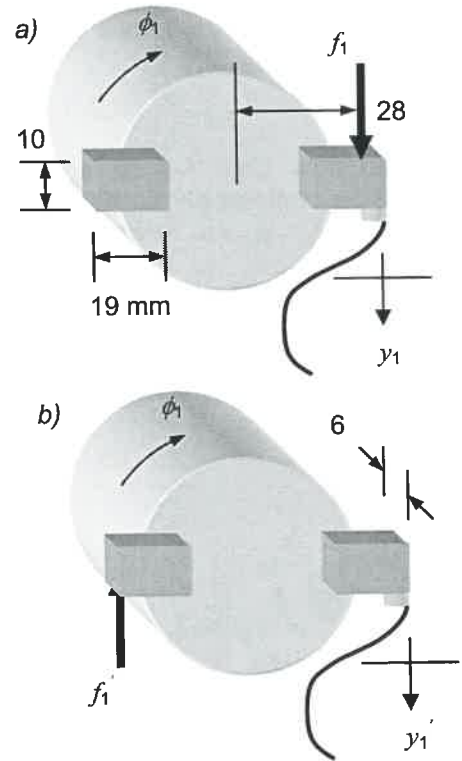


FIGURE 6. a) First torsional measurement – the lateral force f_1 was applied and the y_1 lateral response was measured; b) second measurement – the lateral force f_1' was applied in the opposite direction and the y_1' lateral response was measured.

FREE-FREE BEAM RESULTS

Comparisons between receptance predictions and experimental results for a 1056 mm long 6061-T651 aluminum stepped diameter beam (see Fig. 5) are presented. The beam material properties were assumed to be: $E = 70$ GPa, $\nu = 0.33$ (Poisson's ratio), $G = \frac{E}{2(1+\nu)} = 26.3$ GPa, $\rho = 2700$ kg/m³, and solid damping factors $\eta_G = 0.0003$, and $\eta_E = 0.0001$, which were used to add damping to the component receptances by multiplying the shear modulus by $(1+i\eta_G)$ and the elastic modulus by $(1+i\eta_E)$. Impact testing was applied to identify the beam receptances. A measurement bandwidth of 5000 Hz was applied, although the impact input energy above 3000 Hz was minimal (nylon hammer tip).

While the axial measurements were straightforward to complete, determining the torsional receptances posed a more significant challenge. As shown in Fig. 6, two aluminum

measurement tabs were fixed in a symmetric pattern to the beam end using cyanoacrylate. In a first measurement, the hammer impact and acceleration measurement were completed on the same tab (Fig. 6a). Because the impact was applied at a distance of 28 mm from the beam center, a torque was introduced. This excited the torsional vibration modes. However, the lateral force also effectively excited the bending modes and “corrupted” the torsional receptance. To cancel the bending modes, a second measurement was completed. The accelerometer location was not changed, but the force was applied on the opposite tab at the same radius, but in the opposite direction (Fig. 6b). This caused the sign of the bending modes to reverse, but the torque direction was unchanged so the torsional response was nominally the same. To reject the bending modes and isolate the torsional modes, the two measurement results were averaged on a frequency-by-frequency basis. In order to convert from the averaged acceleration-to-force (inertance) data to rotation-to-torque data, two steps were required. First, the inertance was converted to receptance (displacement-to-force) by dividing by $-\omega^2$. Second, the torsional response was obtained from this result using Eq. 18, where r is the radius for the force application and response measurement (28 mm in Fig. 6).

$$\frac{\phi}{t}(\omega) = \frac{\frac{y}{r}}{f \cdot r} = \frac{y}{f}(\omega) \cdot \frac{1}{r^2} \quad (18)$$

For assembly prediction, the three free-free, constant diameter sections from Fig. 5 were rigidly coupled. First, the left and middle components were coupled. Second, this subassembly was coupled to the right component to give the assembly receptances. Comparisons between measurement and prediction are given in Figs. 7 (torsional) and 8 (axial).

The measured and predicted torsional natural frequencies from Fig. 7 are {901, 3552, and 4462} Hz and {908, 3531, and 4440} Hz, respectively. This corresponding percent error between measurement and prediction is {-0.8, 0.6, and 0.5} %. It is proposed that the anti-resonant frequency offsets are the result of the averaging step. For the axial receptances in Fig. 8, the percent error between the measured and

predicted natural frequencies, 1912 Hz and 1937 Hz, respectively, is -1.3%.

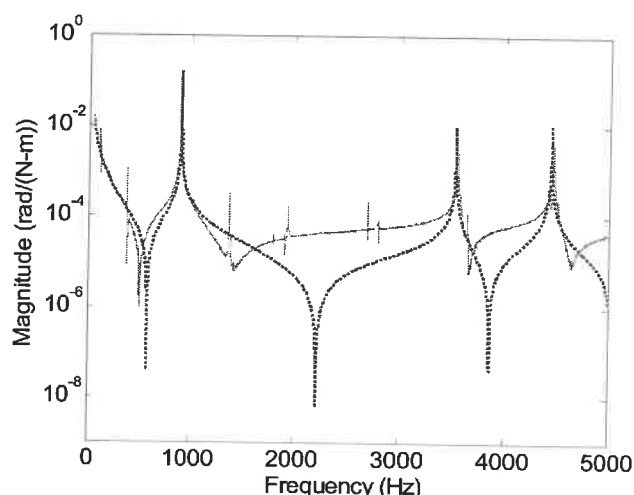


FIGURE 7: Comparison between predicted (dotted) and measured (solid) torsional receptances for stepped diameter beam.

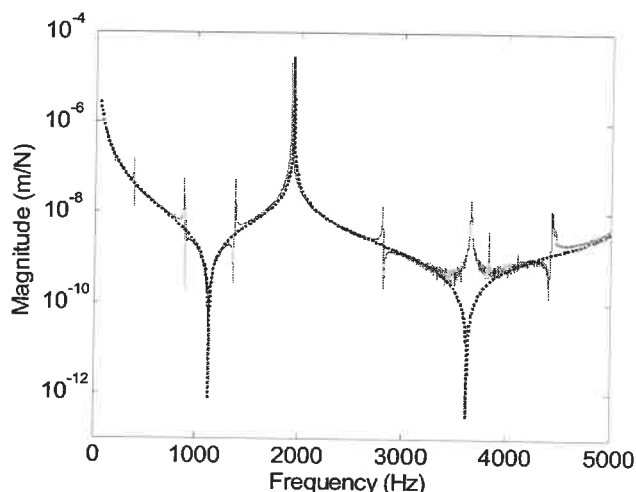


FIGURE 8: Comparison between predicted (dotted) and measured (solid) axial receptances for stepped diameter beam.

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REDUCTION IN SURFACE ROUGHNESS OF HIGH CARBON STEEL USING PULSE ELECTROCHEMICAL MACHINING

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1. INTRODUCTION

The polishing of metals improves the engineering as well as aesthetic aspects of a workpiece. Polishing removes stains, surface corrosion, and scratches are removed to create a smoother, more lustrous, finish. The lower roughness results in lesser friction – an important consideration for bearing materials. Traditional metal polishing using abrasives can be tedious especially with harder work materials. Nontraditional processes such as electrochemical machining (ECM) can be a viable alternative in such cases.

ECM involves anodic dissolution of the workpiece material. Electrochemical machining provides improved results over traditional polishing processes in that it removes surface stresses caused by mechanical working. It can also remove micro-scratches. Removal of these surface defects leads to improved fatigue resistance [1]. The hardness of the workpiece is of little concern as the tool electrode never makes physical contact with the workpiece. Sodium nitrate is the most commonly used electrolyte used in ECM, whose concentration typically varies in the range of 1 and 5 M [2-5]. ECM also often incorporates an acid electrolyte, the most common being sulfuric acid [6]. Extensive studies have been performed on the electropolishing of metals such as stainless steels and aluminum and titanium alloys [7]. However, the electropolishing of high carbon steels has not yet been studied adequately. It would seem to be an attractive method to improve the appearance and physical characteristics of A2 tool steel with an initial roughness (R_a) of approximately 1.08 μm .

An experimental investigation into the feasibility of polishing A2 tool steel in several different electrolyte solutions is discussed in this paper.

Experiments were conducted on workpieces with different values of machining time, interelectrode gap, current, and voltage. Pulse on and off times were held constant at 1 ms, with a resulting duty factor of 50%. The polished surfaces were then analyzed both visually and mechanically to determine surface roughness.

2. EXPERIMENTAL DETAILS

An experimental setup has been constructed to perform the necessary experiments in this investigation. The setup consists of an electrolyte holding tank (~1 L), the electrolyte, a workpiece fixture, a vertical slide with stepper motor which acts as a tool holder and for setting the interelectrode gap, stepper motor controller, pulse/D.C. power supply, relay plugged into digital timer, and personal computer with USB data acquisition device. Experimental setup is shown in Figure 1. Experiments incorporated combinations of NaNO_3 and H_2SO_4 at ambient temperature.

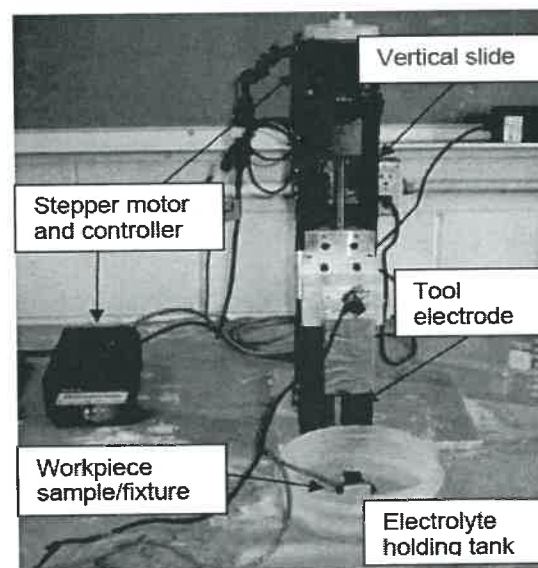


FIGURE 1. Electropolishing Setup [8]

The final machined surface roughness is measured by a WYKO Roughness/Step Tester (RST) system. This laser interferometric microscope system is capable of accurately measuring surface roughness down to 3 to 4 angstroms [9]. A series of 2^k factor screening experiments, where each factor is tested with a high value and a low value, were conducted to determine each factor's significance. The factors and their levels considered in these experiments are summarized in Table 1.

TABLE 1. Experimental Parameters

Voltage	5 V, 25 V
Current	100 A, 200 A
Interelectrode Gap	50 μm , 500 μm
Machining Time	1 min, 10 min
Electrolyte	
NaNO ₃	1 M, 5 M
H ₂ SO ₄	0.1 M, 0.05 M
NaNO ₃ /H ₂ SO ₄	1M, 5 M/0.1 M, 0.05 M

3. RESULTS AND DISCUSSION

The first set of experiments were performed with a purely neutral salt solution of NaNO₃ in either 1 M or 5 M concentration. Thirty-two experiments were conducted (2^5) which considered all combinations of factors and factor levels, the response variable being the surface roughness R_a in μm . An analysis of variance (ANOVA) was conducted with this data, only considering the main factors, with the response variable being surface roughness R_a . The ANOVA table is shown in Table 2. With an $\alpha = 0.05$, only the voltage was found to be statistically significant. The response variable, surface roughness, had a mean of 0.78 μm and a standard deviation of 0.29.

An observation that seemed common to all of these experiments was that the electrolyte often had a dark brown/orange color after machining. This discoloration is most likely due to the presence of metal hydroxides. The electrolyte was often warm as well. Short circuits were common problem that required termination of several experiments early. It is possible that these short circuits may have actually damaged the surface and increased its roughness.

It was also observed that the workpieces often became darkened by the experiment. This darkening would sometimes occur at the place where material removal took place, sometimes at areas away from the machining location. Over time, these samples also seemed more susceptible to rusting or oxidation.

The second set of experiments incorporated a purely acid electrolyte solution of H₂SO₄. Each experiment would use either 0.1 M or 0.05 M. Though most commonly used in 0.1 M concentration, a lower concentration would be used as well. A total of 32 experiments were conducted.

An ANOVA of these experiments, shown in Table 3, found no factors to be statistically significant when using an $\alpha = 0.05$. Surface roughness had a mean of 0.68 μm and a standard deviation of 0.20.

In observing results of these experiments, it was found that discoloration of the acid electrolyte did not occur as it had with the salt solution - it remained relatively clear. The electrolyte was often warm, though, at the end. Short circuiting did occur, but not nearly as often as in the previous experiments. Surface darkening also did not seem to be as much of a problem as had previously occurred.

Another common electrolyte in ECM is an acid/salt mixture of varying concentrations. The third set of experiments considered an electrolyte composed of two different values of both acid and salt. Acids were again 0.1 M or 0.05 M, and NaNO₃ 1 M or 5 M. Interelectrode gap was held constant at 250 μm .

As in the previous set of experiments, an ANOVA with $\alpha = 0.05$ found no factors to be statistically significant. Mean surface roughness was 0.60 μm with a standard deviation of 0.16. ANOVA table shown in Table 4.

Surface darkening was again a problem as it had been in the salt solution experiments. Short circuiting was also quite common. There did not seem to be the problem, though, of further oxidation of workpiece surfaces after machining.

The final appearance of the electrolyte more resembled the acid solution experiments. It was clear with no brown/orange color evident.

TABLE 2. ANOVA for Neutral Salt Electrolyte

Dependent Variable: RA

Source	Type III Sum of Squares	df	Mean Square	F	Sig.
Corrected Model	.996 ^a	5	.199	3.395	.017
Intercept	19.369	1	19.369	330.178	.000
MACHTIME	.243	1	.243	4.147	.052
CONC	6.682E-02	1	6.682E-02	1.139	.296
IEG	9.028E-02	1	9.028E-02	1.539	.226
CURRENT	2.004E-02	1	2.004E-02	.342	.564
VOLTAGE	.575	1	.575	9.806	.004
Error	1.525	26	5.866E-02		
Total	21.889	32			
Corrected Total	2.521	31			

a. R Squared = .395 (Adjusted R Squared = .279)

TABLE 3. ANOVA for Acid Electrolyte

Dependent Variable: RA

Source	Type III Sum of Squares	df	Mean Square	F	Sig.
Corrected Model	9.719E-02 ^a	5	1.944E-02	.435	.820
Intercept	14.797	1	14.797	331.464	.000
MACHTIME	1.004E-02	1	1.004E-02	.225	.639
ACIDCONC	1.103E-02	1	1.103E-02	.247	.623
IEG	5.085E-02	1	5.085E-02	1.139	.296
CURRENT	2.769E-04	1	2.769E-04	.006	.938
VOLTAGE	2.500E-02	1	2.500E-02	.560	.461
Error	1.161	26	4.464E-02		
Total	16.054	32			
Corrected Total	1.258	31			

a. R Squared = .077 (Adjusted R Squared = -.100)

TABLE 4. ANOVA for Acid/Salt Electrolyte

Dependent Variable: RA

Source	Type III Sum of Squares	df	Mean Square	F	Sig.
Corrected Model	.118 ^a	5	2.365E-02	.955	.463
Intercept	11.454	1	11.454	462.431	.000
MACHTIME	1.810E-02	1	1.810E-02	.731	.400
ACIDCONC	1.498E-02	1	1.498E-02	.605	.444
SALTCONC	2.425E-02	1	2.425E-02	.979	.332
CURRENT	2.053E-05	1	2.053E-05	.001	.977
VOLTAGE	6.089E-02	1	6.089E-02	2.458	.129
Error	.644	26	2.477E-02		
Total	12.216	32			
Corrected Total	.762	31			

a. R Squared = .155 (Adjusted R Squared = -.007)

The three sets of initial experiments provided several interesting results. The first of these was that only one factor (voltage), in the first set of experiments, was found to be statistically significant. This indicates a need to conduct a more comprehensive experimental study with a wide range of parameter levels and replicates. Another experimental observation is that short circuiting was a very common problem when dealing with electrolytes containing NaNO_3 . The problem was less so in the $\text{NaNO}_3/\text{H}_2\text{SO}_4$ electrolyte compositions. It would seem that the presence of the acid forces the dissolved metal ions to remain in solution rather than precipitate out as metal hydroxide, discoloring the solution. Experimental results for the three experimental data sets are summarized in Table 5.

TABLE 5. Experimental Results

Experimental Set	Electrolyte Composition	Average R_a
1	NaNO_3	0.78 μm
2	H_2SO_4	0.68 μm
3	$\text{NaNO}_3/\text{H}_2\text{SO}_4$	0.60 μm

In selecting the best electrolyte, it was determined that the pure acid electrolyte would be the one employed for subsequent experiments. Even though its average roughness was in the middle of the three compositions, the lesser darkening of workpiece surfaces and reduced chances of short circuiting makes it a better choice. It is found that the purely acidic (H_2SO_4) electrolyte gives smoother surfaces over NaNO_3 and $\text{NaNO}_3/\text{acid}$ electrolyte combinations. An acid concentration of 0.1 M with an interelectrode gap of 500 μm and power supply settings of 100% current and 25 volts provided a lustrous surface. These results highlight the potential of electrochemical machining as an alternative polishing method for high carbon steel. The pure acid electrolyte has shown that it alone can be implemented in the electropolishing process.

ACKNOWLEDGEMENTS

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MICROMILLING OF MOLDS FOR MICROFLUIDIC BLOOD DIAGNOSTIC DEVICES

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INTRODUCTION

In the application to diagnostic devices for blood flow, microfluidic devices with high aspect ratios (such as features with lateral and out-of-plane dimensions on the order of 250-1000 μm) are particularly useful as they enable the testing of a spectrum of flow conditions (e.g., shear rates) while operating at the limiting dimensions for physiological relevance in blood rheology experiments. However, such features are difficult and time consuming to fabricate with conventional semiconductor processes such as lithography, deposition, or etching. Further, structures with multiple out-of-plane dimensions are tedious, expensive, or impossible to fabricate with these semiconductor processes. Alternative micromanufacture methods which are capable of producing high aspect ratios, such as micro-edm and micro-ecm, require the manufacture of initial prototypes to serve as customized electrodes [1].

Thus 3-D microfabrication techniques that can augment or supplant these current methods have great value. In this work, they can enable biomedical instrumentation with applications in both research and clinical settings for examining shear related effects of blood clotting and their relationship to high-throughput pharmaceutical assays.

Micromilling is uniquely suited to mold-making because of its flexibility. High aspect ratios up to 10 to 1 are possible [2, 3], and multiple out-of-plane dimensions are easily achievable on a 3-axis machine. In this abstract, we present a 3-axis micro/meso-scale milling machine and its application to the fabrication of an aluminum mold to cast polymer-based microchannel devices assaying blood clot formation under varying shear rates. Experimental validation of the performance of the flow channels as well as evaluation of clot formation is also presented.

METHODS AND RESULTS

We constructed a customized micromilling machine by retrofitting a standard Sherline, Inc. mini milling machine model 5400. The machine was retrofit with high-torque brushless DC motors from Motion Control Group, Inc. in order to allow the high accelerations (1.6 g) necessary to accurately produce high-curvature features at high feedrates. The motors were fit with high-resolution 20,000 count/rev absolute optical encoders, providing a 0.05 μm maximum encoder resolution. The standard spindle was replaced with an 80,000 rpm electric spindle from NSK, Inc., in order to increase cutting speeds with the small tools and allow for increased feedrates while maintaining a reasonable chipload.

A control system for the machine was created using NI control hardware and the LabWindows development software. An NI 7350 motion control card was employed. This hardware allows for a PID update rate of 125 μs for control of three axes simultaneously. The control system software was designed with basic machine tool control capabilities including closed-loop linear positioning, velocity-controlled contouring, monitoring feedback, and G-Code emulation.

The control system was also designed to enhance the accuracy of the micromilling machine. PID feedback was tuned for reduction of velocity oscillation and zero steady-state positioning error. A backlash compensation algorithm was added to the positioning control to reduce or eliminate error in the leadscrew. In order to improve the depth accuracy, a conductive touch-off method with accuracy within 1 μm [4] was implemented. A summary of the micromilling machine is shown below in TABLE 1.

TABLE 1: Micromilling machine characteristics

Parameter	Value
Spindle speed	80,000 rpm
Encoder resolution	0.05 μm
Maximum acceleration	1.6 g
Maximum velocity	100 mm/s
Touch-off accuracy	<1 μm
PID update rate	125 μs
Work volume	200 x 150 x 100 mm

The mold geometry was designed for the manufacture of channel geometries to produce a range of shear rates on blood across the pathologically relevant range of 4300-7100 s^{-1} [5]. It was then milled using square endmills ranging from 150-1000 μm in diameter. The larger tools have a standard flute length equal to 3x the diameter, while the small tools have been necked to 10x the diameter in order to allow cutting of the small features to the depth required by the mold. This ability to achieve high aspect ratios on small features is a key advantage of the micromilling method.

The mold was cut using feedrates between 0.1 and 10 mm/s, and axial depths of cut ranging from 10-100 μm . Average R_a surface roughness was measured by a scanning white-light interferometer (Zygo NewView 200) as 150 nm on the device's critical planar faces. Total machining time was approximately 30 min, including tool change time. The custom micromill is shown below in FIGURE 1.

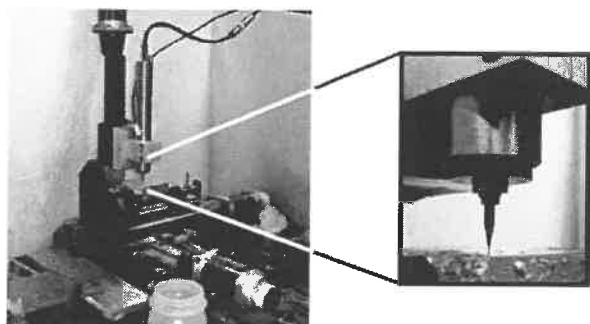


FIGURE 1. The customized micromill discussed in this work (left) and 1000 μm diameter endmill (inset).

Next, the micromilled mold was used to cast a layer of transparent, high fidelity, elastomeric polymer PDMS. The polymer layer was then bonded to a glass slide to create enclosed

channels and coated with Type I collagen to initiate clot formation. Next, blood was flowed through the resultant device at a fixed pressure to exert the desired shear rates on the blood samples and form resultant blood clots. These shear rates were verified by Poiseuille flow based calculations, bulk flow rate tests, and by particle velocimetry. The mold and cast PDMS device are shown below in FIGURE 2.

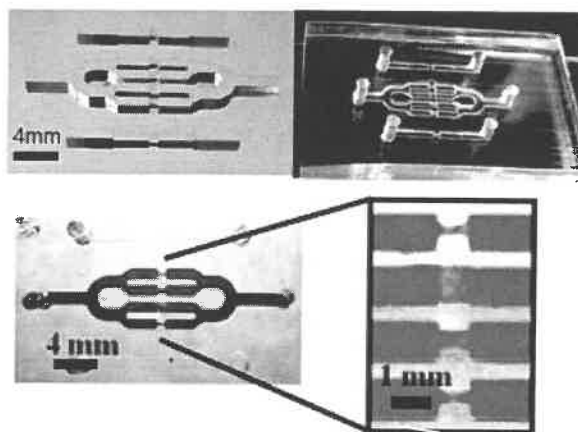


FIGURE 2. Micromilled aluminum mold (above left), cast PDMS device (above right). Device after flow experiment using porcine blood (below) showing clots formed within the microchannels (inset).

Volumetric clot formation was evaluated by image capture in x and y over 12 μm intervals for the height of the channel. Thrombus volume was then evaluated as the sum of the intensity, $\Sigma I(x,y,z)$, and volumetric growth rate was quantified as the percent of intensity increase in three dimensions over time, $\Sigma I(x,y,z,t_i) / \Sigma I(x,y,z,t_{\text{final}})$ for a given time t_i , where t_{final} is the time at which complete channel occlusion by the formed clot occurs. Initial results from these experiments have shown the three characteristic phases of clot growth previously documented in the literature: adhesion (initiation of clot), acute growth, and occlusion (when flow in the channel is completely stopped by the clot) [6]. Experimental results of one such assay are shown below in FIGURE 3.

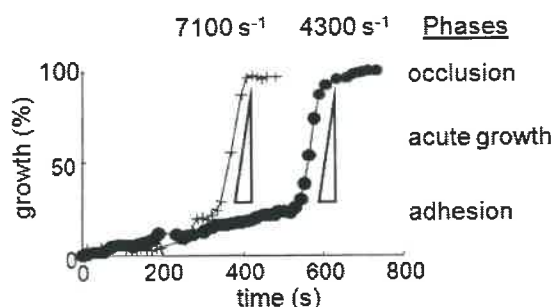


FIGURE 3. Clot growth from whole blood subjected to two different shear rates. Growth (%) is $\Sigma I(x,y,z,t_i) / \Sigma I(x,y,z,t_f)$ for time t_i and occurs in three distinct phases of adhesion, acute growth, and occlusion

DISCUSSION

This micro-manufacturing method presents a significant improvement over traditional soft lithography semiconductor fabrication processes [7] for this application since the latter is unable to achieve the required range of feature heights and widths from 250 to 1000 μm at aspect ratios from 1:1 to 1:4. Although we report a 30-minute manufacturing time required for the mold, this amount of time may be much higher than is necessary. Much of the time required to cut the part was due to the low feedrates achieved by the smaller tools. Low feedrates were necessary not primarily because of the size of the tool, but because of the control method employed on features of high curvature in order to limit the contouring error. Furthermore, although the extremely low surface roughness possible with semiconductor techniques is not achievable in milling, we have found that the finish is sufficient for this application.

Casting PDMS devices from a robust aluminum mold allows for fast, affordable, and reproducible device manufacture compared with other current 3-D micro-manufacturing methods. The low surface roughness produced from this micromilling method also allowed for both easy polymer release from the mold without the aid of additional chemical agents as well as smooth bonding of the PDMS polymer to glass.

The device itself presents a low cost, high throughput, alternative method of assaying the clotting response of blood to varying flow conditions and has shown to be amenable to optical acquisition and analysis.

CONCLUSIONS

In this work we have described the development of a micromilling machine, its associated control apparatus, and its application to the production of polymer-based microfluidic devices for studying the process of blood clotting under varying flow conditions. With our micromilling method, we were able to produce a mold with a wide range of feature heights and widths unachievable by semiconductor fabrication methods. Future work on the micromilling machine includes modification of the control method to increase the feedrates achievable on small features, to reduce the mold machining time. Blood assay devices will continue to be developed toward pharmaceutical assays and biomedical research.

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VIBRATION ISOLATION SYSTEM WITH HIGH ACCURATE POSITIONING TECHNOLOGY OF SEMICONDUCTOR LITHOGRAPH MACHINE

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INTRODUCTION

In the field of accurate positioning system, an active vibration system is generally used. Especially it is indispensable unit in the system which is required a highly accurate isolation performance from outside, such as floor vibration, in the semiconductor lithograph machine[1]. Isolation performance from floor vibration to the body is particularly called transmissibility. There are many development elements in the semiconductor lithograph machine. However, that are assumed the introduction only of the title(Incline compensation of isolated table using voice coil motor[2], Wide bandwidth by using multi VCM for isolation unit, Improvement of compensation for accelerometer and high bandwidth for isolation unit[3], etc) by limiting space.

In recent years, the progress of semiconductor is remarkable development, therefore outside vibration isolation performance become more important factor than current machine for total machine accuracy. Isolation vibration system is generally supported by air-spring[4].

In this paper, we will introduce two methods of improvement of isolation performance, that is, passive and active method.

At first, as passive method, there is one of mechanical method in order to increase vibration rejection ratio, which is reduced the stiffness of air-spring by using auxiliary tank, Helmholtz resonance is caused by using auxiliary tank, and vibration rejection ratio become worse. In usual as is introduced in the Handbook[4], we can reduce the peak of Helmholtz-resonance by adjusting the aperture with the built-in the connected pipe. Furthermore there is the best

aperture as is introduced in the Handbook. However, there is no reference regarding the relationship between Helmholtz frequency and vibration rejection ratio.

Secondly, it is as to active method, there is the method by control technique using servo valve. We can achieve the same performance to passive method of auxiliary tank by means of servo valve control technique, it is called here as "pseudo auxiliary tank".

ISOLATION DEVICE AND IMPLEMENTATION OF MAKING LOW NATURAL FREQUENCY METHOD

Structure

Figure 1 shows an active vibration unit in the semiconductor lithograph machine. The semiconductor exposure machine sets up on the installation floor that is called a pedestal which is in the base of the main body of this machine. Voice coil motor whose main body of the exposure machine is an actuator of the machine(hereafter, abbreviation as VCM)and an air-spring are mounted on the base of the main body of the machine. In addition, the projection lens, optics system of the alignment etc. are mounted on the main body of the exposure machine. Thus, this main body is isolated from the floor vibration with the air-spring , and the isolation performance greatly related to the stiffness of the air-spring, that is , the capacity of the volume of the air-spring. In general, main body is supported on the pedestal through three air-spring support legs. In each supporting legs of the machine, it provides with three sensors that measure a pedestal and a relative position

with the main body respectively in the vertical direction. In addition, to measure the position of the main body from the floor, there are 3(6 in total with vertical direction) positioning sensors for horizontal direction. The position sensing device of the main body from the pedestal are adjacent and arranged with three air-springs. In similarly, three accelerometers are mounted in the vertical direction, another three are mounted horizontally, and the movement of six degrees-of-freedom is measured. Moreover three VCM are horizontally and vertically arranged.

Means of making low natural frequency

There is a demand that it wants to decrease the natural frequency of the main body supported with the air-spring. It is easy to adjust the natural frequency of the main body to 1-3Hz equipped with the air-spring. Since specifications of the machine came to become severe, further isolation performance improvement has been imposed on the isolation machine.

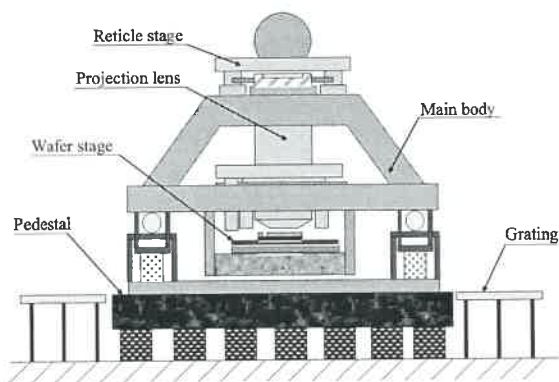


FIGURE 1. Semiconductor lithograph machine apparatus

In this paper, we introduce the method with a auxiliary tank as one of the making to a low natural frequency[5]. The air room of the air-spring is connected with auxiliary tank by piping, and air-spring stiffness can be decreased because of the increase of the capacity of the air room. This design requirement is written in the reference [4]. Concretely, referring to Figure 2, it is explained that the best diameter that puts out the maximum damping force exists for the squeezing diameter mounted in the model which is composed of the volume of main tank, the volume of auxiliary tank, and the piping part when the auxiliary tank is connected with the main tank by piping. In addition, it is said that the larger the squeezing attenuation is the capacity

of the auxiliary tank, the more effective. However, it became clear during the study of the

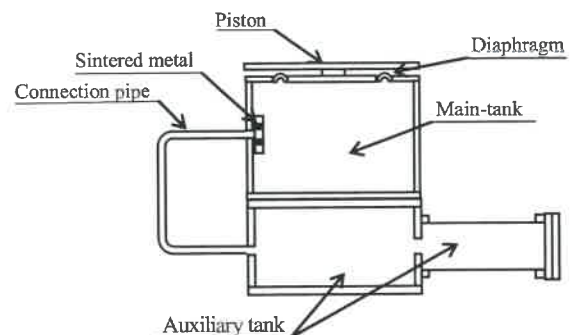


FIGURE 2. Main tank connected auxiliary tank apparatus

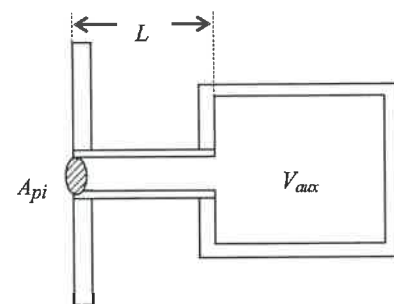


FIGURE 3. Auxiliary air tank simple model

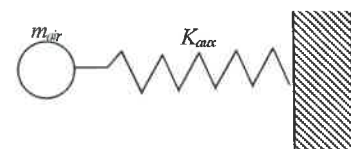


FIGURE 4. Mass-spring simple model

authors that critical relationship are actually in the piping length between the Helmholtz resonance frequency and isolation performance.

Mechanism of Helmholtz resonance

The modeling when two auxiliary tanks are connected with the main tank by piping is expressed as the model of Figure 2. At first, frequency f_0 Hz of the Helmholtz resonance is the following expression for Figure 3 where only one auxiliary tank was connected by piping.

$$f_0 = \frac{C}{2\pi} \sqrt{\frac{A_{pi}}{LV_{aux}}} \quad (1)$$

Here, C : speed of sound m/s and A_{pi} : piping part cross sectional area m^2 and L : piping part length V_{aux} : volume of auxiliary tank m^3 . Moreover the mechanical model in Figure 3 is expressible as shown in Figure 4. In addition, the mechanical model of three inertia system is requested from the supporting leg of Figure 5 where two auxiliary tanks were connected with the main tank, it becomes Figure 6. That is, it is a mechanical coupling vibration of 1DOF. According to the reference [4], air-spring stiffness K can be written as equation (2)

$$K = \frac{dW}{dZ} = A \frac{dP}{dZ} + P \frac{dA}{dZ} = \gamma \frac{(P + P_a) A^2}{V} \quad (2)$$

Here P_a : atmospheric pressure and Z : compression deformation from a neutral position of the air-spring. Using equation (2) and based on the mechanical model of Figure 6, when isolation performance z/z_0 from displacement z_0 of the floor vibration to displacement z of M in mass is simulated, it becomes Figure 7.

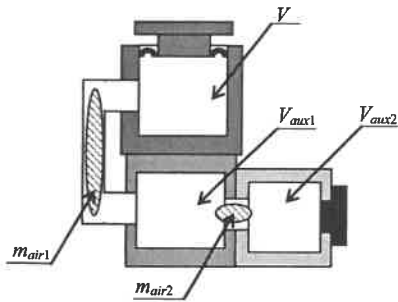


FIGURE 5. Main tank connected two auxiliary tanks

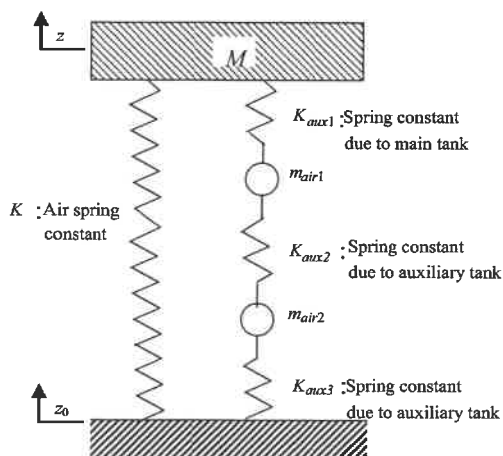


FIGURE 6. Mechanical model for FIGURE 5.

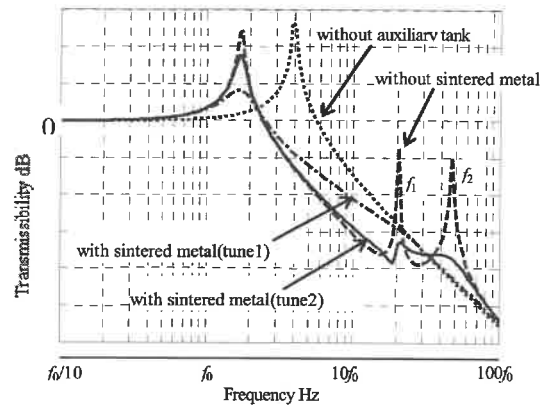


FIGURE 7. Comparison between with and without sintered metal

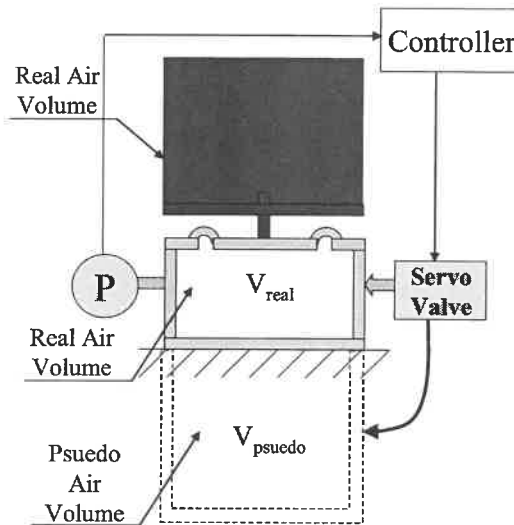


FIGURE 8. Overview of pseudo tank by using servo valve

The Helmholtz resonance frequency appears in the low frequency region by having connected it with the auxiliary tank by piping as understood in Figure 7. In addition, the most important things is not to achieve the effect of the auxiliary tank in the high frequency region according to this resonant frequency. Therefore, it means isolation performance can not reach the objective.

Pseudo auxiliary tank method

In the above mentioned, we explained the active method of improving the isolation

performance with a mechanical auxiliary tank. Here, in addition, we explain the passive method of achieving the effect similar to the auxiliary tank by the control with servo valve. We study by easy model of one degree-of-freedom shown in Figure 8, where P_s : pressure sensor, M : body mass. This unit is composed of mass, air-spring, servo-valve, and controller.

Air-spring stiffness is written as next expression,

$$K = \frac{\gamma A^2 P}{V} \quad (3)$$

where K : stiffness of air-spring, V : volume of air-spring, A : actual effective area of air-spring, P : neutral pressure, γ : polytropic coefficient. To become pressure change similar to a big volume, the controller puts air in and out with a pressure sensor and servo-valve. In other words, pressure can be changed effectively in the main tank of V_{real} so that the volume may become a pressure change similar to the pressure change of $V_{real} + V_{psuedo}$.

Figure 9 shows the control block, according to this figure, it is possible to achieve the objective/specification by feed back the differentiation of pressure signal.

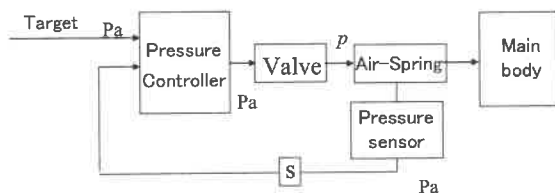


FIGURE 9. Overview of pseudo tank by using servo valve

$$p = \frac{G_q}{\left\{ \beta_0 \left(V_0 + \frac{G_q k_{gv}}{\beta_0} \right) \right\} s + c} \cdot \frac{Ms^2 + Ds + K}{Ms^2 Ds + K + \frac{A_0^2 s}{\left\{ \beta_0 \left(V_0 + \frac{G_q k_{gv}}{\beta_0} \right) \right\} s + c}}$$

Here, p is a target pressure, and β_0 : compressibility, V_0 : volume in air tank, G_p : flow rate gain, k_{gv} : differential feed back gain, A_0 : effective area, M : body mass, D : air-spring damping, K : air-spring stiffness

Finally, we will introduce the comparison when with or without this algorithm. There is a pressure response characteristic in Figure 10.

In Figure10, the line shown in green is original data. On the other hand, the line shown in blue shows the effect when this algorithm is applied.

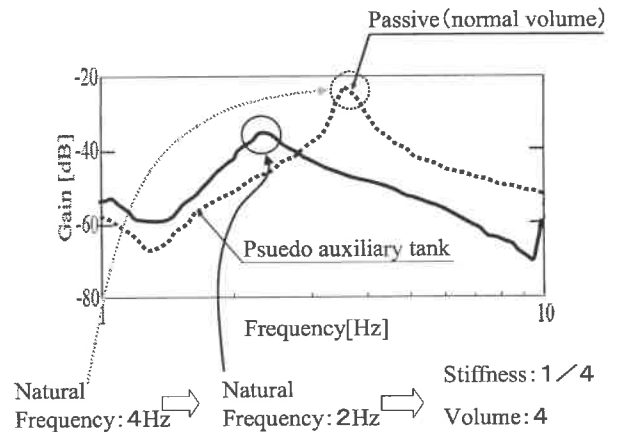


FIGURE 10. Comparison effectiveness of with/without this method

Since it is half at the natural frequency, it can be said that the stiffness is one-quarter. that is the experimental result is proof of effective.

CONCLUSIONS

The effect of the auxiliary tank to increase isolation performance is a band up to Helmholtz resonance frequency. The auxiliary tank did not function as an air volume more than this frequency, and isolation performance becomes only air volume of main tank. The point of the design is an effect of damping. In a word, it is necessary to set the hole size of the porous quality that doesn't deprave isolation performance below the Helmholtz resonance frequency and suppresses the resonance peak of the Helmholtz resonance. Furthermore, the air volume effect of the auxiliary tank is able to be pseudo produced with a servo valve in the main tank.

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DETERMINING TOOL DIAMETER FOR MACHINING OF SURFACE TEXTURE

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1. INTRODUCTION

Machining of surface texture is carried out in the manufacturing process of products such as automobiles and electronic goods to improve the touch and visual effects of product surfaces[1]. During the machining of surface texture, machining time and accuracy depend greatly on the diameter of the used ball end mill. However, there sometimes exists no appropriate tool diameter for the machining of the surfaces with pits because surface texture is composed of small asperities. In addition, the method of deciding the appropriate tool diameter can be difficult because small tool diameters need to be used. On the other hand, observations of the surface texture features show that areas around grooves at the course line, that is, at the convex part, are concave, so the surface curvature radius is small. In this study, tool diameter was calculated based on the curvature of the textured surface. The whole area of the surface was divided into machinable and non-machinable areas, and the tool diameter was gradually changed according to these areas. The tool path used in this study was assumed to be a spiral that centers on the top of a convex shape.

2. CREATION OF SURFACE TEXTURE

Figure 1 shows the procedure of creating surface textures consisting of the following steps.

- (a) The designer creates the surface texture of a narrowness area where several patterns can be identified.
- (b) Same shapes existing within the designed surface texture are classified according to the convex shape type[2].
- (c) CL data on narrow areas is generated for each convex shape type.
- (d) CL data on wide areas is generated by joining the above CL data by moving or rotating.
- (e) The CL data on wide areas is mapped to the plane or geometry.
- (f) The surface texture is machined by milling based on the generated CL data on wide areas.

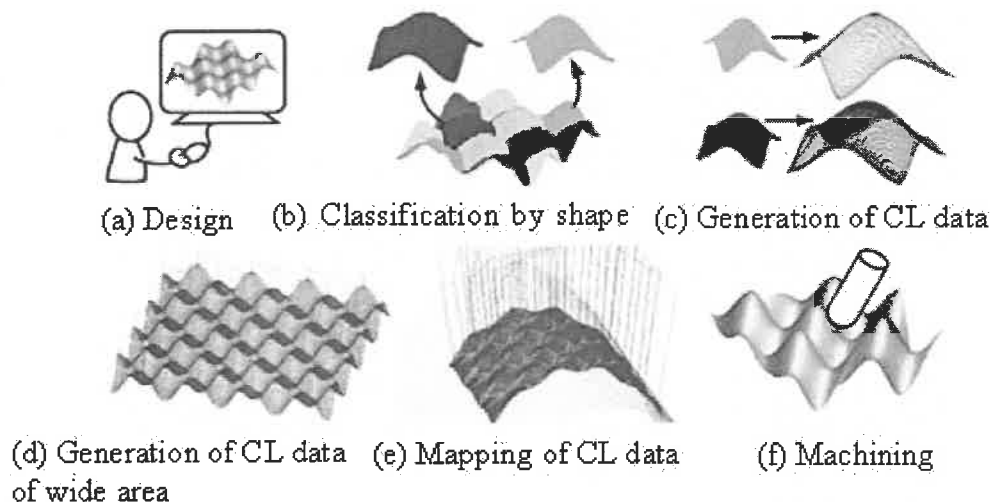


FIGURE 1. Creation of Surface Texture

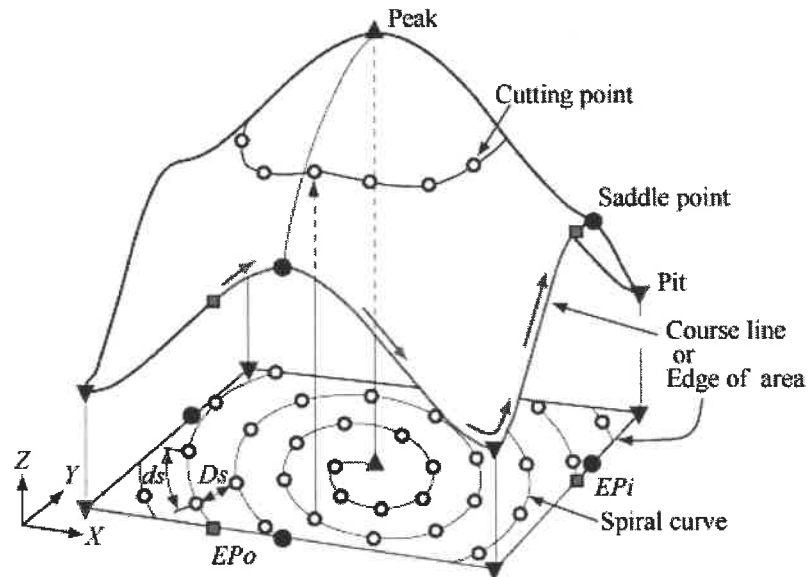


FIGURE 2. Generation of CL Data

3. GENERATION OF CL DATA

As described above, CL data is created for each shape. As shown in Figure 2, a peak is defined as the highest point in its vicinity and a pit is defined as the lowest point in its vicinity. Moreover, the saddle point is defined as a point sandwiched by a ridge and valley. The course line is defined as a line connecting the points from the saddle point to the pit point with the maximum slope. And, the area enclosed by the course line is assumed to be an analytical area.

In this study, the intended tool path was assumed to be a spiral curve centering around the convex part. An Archimedes' Spiral centering on the peak was created for the X and Y planes, on conditions that the spiral distance is D_s and the data distance is ds . The cutting point was calculated by transcribing the data point to the convex part. Cutting points EPO and EPI in the analytical area were calculated respectively if the spiral curve was crossed by course lines or data boundaries as shown in the figure. The cutting point between the EPO - EPI is taken to pass the data points of edges on the course line or data boundary[3].

4. DETERMINING METHOD OF TOOL DIAMETER

The tool diameter is calculated based on the curvature of a curved surface shape[4]. Surface curvature for deciding tool diameter is calculated using equation (1) shown below at each node composing the STL(Standard Triangulated Language) data. Figure 3 shows the method for calculating curvature.

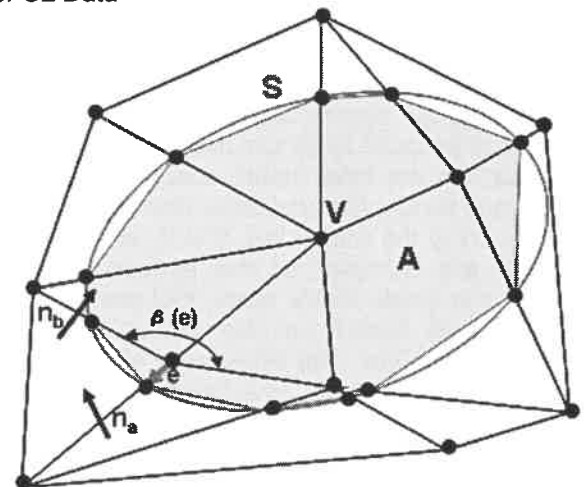


FIGURE 3. Calculation of Curvature

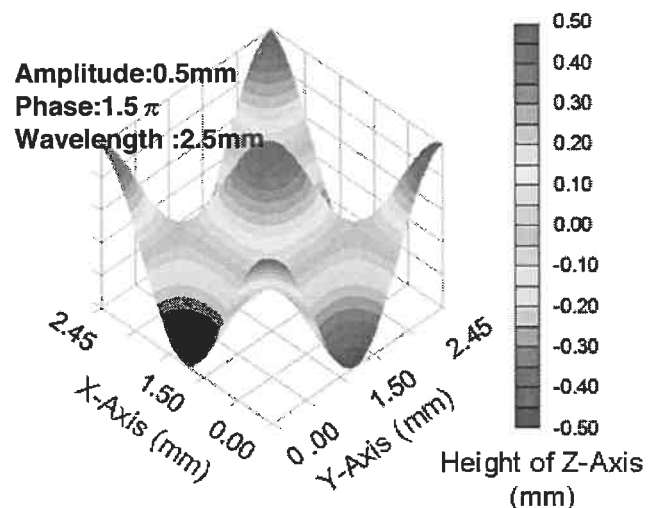


FIGURE 4. Sinusoidal

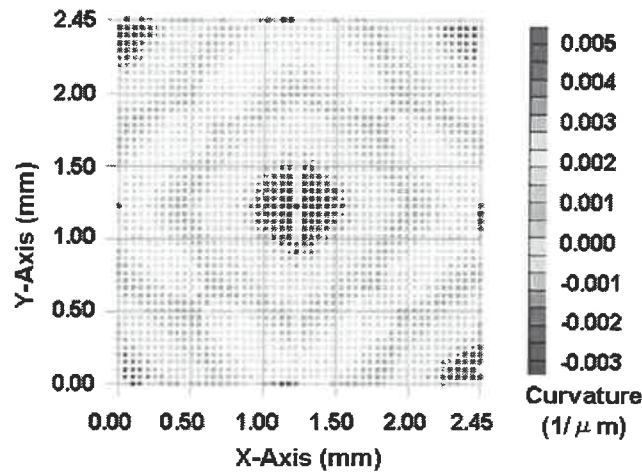


FIGURE 5. Distribution Map of Curvature

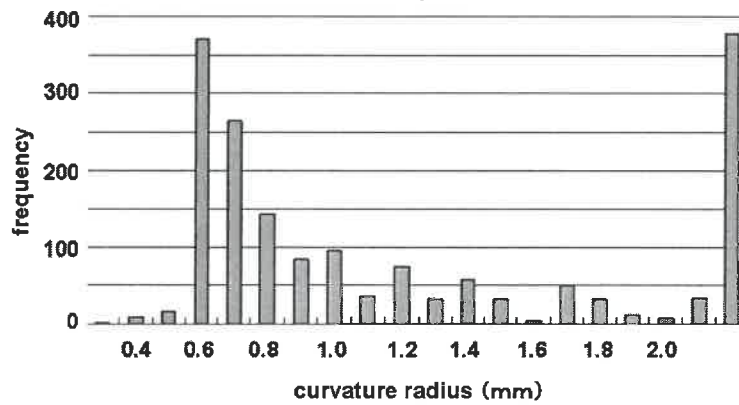


FIGURE 6. Histogram of Curvature Radius

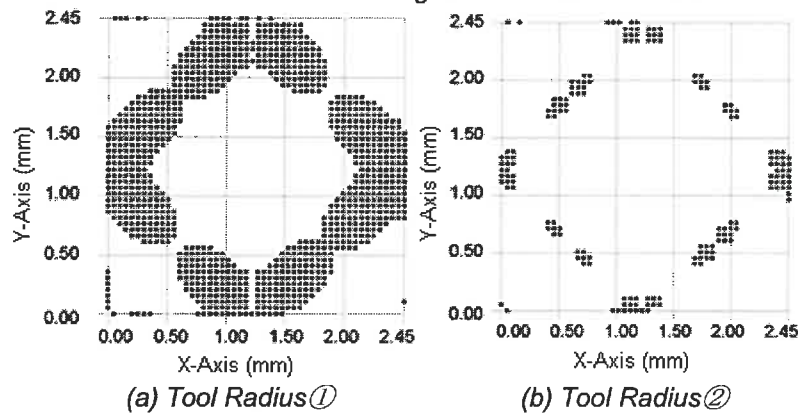


FIGURE 7. Interaction Region

$$T(v) = \frac{1}{|A|} \sum_{\text{edges}} \beta(e) \|e \cap A\| \frac{ee^T}{\|e\|^2} \quad (1)$$

First, taking the vertex whose curvature is to be obtained to be V , a virtual sphere S was created centering around V , and the inside of the sphere was taken to be the area influencing V . $|A|$ is the surface area, e is an edge vector, and $\beta(e)$ is the angle between n_a and n_b that is the normals to the surface connected by e .

In the curvature, the positive is concave surface, and the negative is the convex surface. This means that the larger the surface curvature, the smaller will the tool diameter required be. As an example, the surface curvature of the sinusoidal was calculated with three parameters; amplitude of 0.5mm, phase of 1.5π , and wavelength of 2.5mm, as shown in Figure 4. Figure 5 shows the distribution map of the calculated curvature. Looking at the curvature

distribution map of the surface, it was found that curvatures near the course line are large, requiring tools with small radius in such areas. The tool diameter required can also be estimated by creating the histogram of the curvature radius as shown in Figure 6. Two tools; radius ① (2.0mm) and radius ② (0.2mm) were prepared according to the variation in the curvature radius, and as a result, the black areas shown in Figure 7 were predicted to interfere. In this way, the interfering areas of the tool with radius ① are less than that with radius ②. For both radiuses, increase/decrease of the interfering areas could be seen based on the course line. Applying the changes in the interfering areas around the course line, a tool radius can be set for each area gradually, to decrease the machined areas using a small diameter tool. This means that the tool path can be reduced by using a small tool diameter. Consequently, as shown in (a) of Figure 8, a small area of the curvature near the top should be machined with a big tool like radius ①. And, as shown in (b) of Figure 8, a large area of the curvature near the course line should be machined with a small tool like radius ②. This time, the amount of uncut area, that is, the cusp height, in ball end mill machining was adjusted to about $5\text{ }\mu\text{m}$, in creating the tool path. This as a result successfully shortened the machining time, realizing efficient machining, as shown in Table 1.

5. CONCLUSION

The following conclusions were obtained by calculating the surface curvature of surface texture, examining textured surface features, deciding the tool diameter, and calculating the machining time.

- (1) It was found that that course line-like concave areas with small curvature radius generated by surface curvature appears on the textured surface.
- (2) Proposed a method to determine tool radius appropriate to surface features according to curvature features on the textured surface.
- (3) The machining time can be shortened using tools appropriate for the surface features of surface textures.

In future studies, the machining accuracy of surface textures will be verified through actually ball end milling experiments.

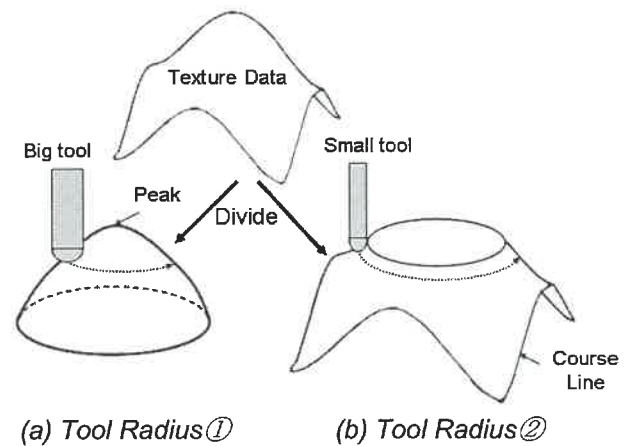


FIGURE 8. Machining Procedure

TABLE1. Machining Time of Hill

Machining method	Existing	New
Feed Rate	600 $\mu\text{m/sec}$	600 $\mu\text{m/sec}$
Tool Radius	R0.2mm	R0.2mm, R2.0mm
Ds	50 μm	50 μm , 150 μm
Machining Time	4minute	2minute 40sec

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DEVELOPING A PREDICTIVE MODEL FOR MILLING STABILITY

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INTRODUCTION

Both physically and economically motivated uncertainties are often present in manufacturing decision situations. To deal with these uncertainties, predictive models can be used. Most methods for predicting functions are statistically based, including regressions and curve fits. However, these methods may be inadequate for the prediction of complex functions, since there is no rigorous method of aggregating information from multiple sources (experimental data, simulation data, and theoretical predictions) and they do not quantify the value of information gathering activities.

Although Bayesian methods for predicting functions can be mathematically complex, they have two major advantages over other approaches. First, any information learned through experiment about the function can be incorporated. This makes these methods more versatile than statistical methods, which are most compatible with simple sets of observables. Second, Bayesian methods enable information to be valued based on its influence on profit, which makes it possible to systematically choose an information gathering scheme that maximizes profit.

Since Bayesian methods have not been widely applied in manufacturing, the focus of this work is to explore their use in this field. To aid in the explanation, stability limit prediction is examined.

BAYESIAN PREDICTION OF FUNCTIONS

These methods are fundamentally different from curve-fitting methods in that they treat a set of predictions rather than a single prediction. Methods which use a single curve as a prediction only answer a single question about an unknown function $g(x)$:

Given an arbitrary knowledge state, what is the most likely $g(x)$?

In contrast, Bayesian methods assign a probability to every function. In other words, for any $g(x)$ the following question can be answered:

Given an arbitrary knowledge state, how likely is it that an arbitrary function = $g(x)$?

The advantage of this approach is that arbitrary information regarding the function of interest can be incorporated into the predictive model. For example, abstract functional assumptions that can't be handled by statistical methods, such as monotonicity, differentiability class, or periodicity, can be incorporated. In addition, multiple informative statements can be aggregated in a mathematically rigorous manner.

Mathematically, probability assignment over a functional space is equivalent to assigning a multivariate distribution over infinite dimensions. This assignment can be simplified by assigning a distribution only over the set of all *measurable quantities*. The most general set of measurable quantities is the set of all functionals, i.e., any operator which takes a function as an input and produces a scalar, such as a definite integral.

BROWNIAN DISTRIBUTIONS

Suppose that the value of $g(x)$ for an arbitrary x depends only on the value of g in a small neighborhood of x . In terms of the conditional probabilities, this is given by:

for any $x_1 < x_2 < x_3 < x_4 < x_5$

$$f_{g(x_3)|g(x_2),g(x_4)} = f_{g(x_3)|g(x_1),g(x_2),g(x_4),g(x_5)} \quad (1)$$

With this assumption, the probability density functional takes the form:

$$f_g(x)(g(x)) = \exp\left(-\int_{-\infty}^{\infty} k(x, g)(\dot{g}(x, g) - \mu(x, g))^2 + c(x, g) dx\right) \quad (2)$$

where $k(x, g)$ is the diffusion field, $\mu(x, g)$ is the drift field, and $c(x, g)$ is the creation and killing field.

Distributions which take this form are referred to as Brownian distributions. Brownian distributions are ideal for predicting and updating when little is known about the function in question. Provided that any information gained through experiment can be expressed in terms of g or its slope at each point, any distribution updated from a Brownian distribution will also be a Brownian distribution.

STABILITY LIMIT PREDICTION USING BROWNIAN DISTRIBUTIONS

To illustrate how Bayesian methods can be used in a manufacturing application, consider the prediction of the speed-dependent stability limit in milling. The stability limit has a particularly complex structure, since it generally has multiple non-differentiable points (Fig. 1). For the numerical example provided here, the true stability limit is generated using the algorithm described in [2]. The input parameters used for the algorithm are summarized in Table 1.

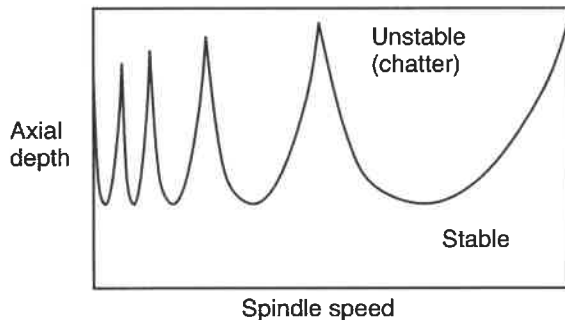


FIGURE 1. Example stability lobe diagram.

TABLE 1. Parameters used to generate the reference stability limit.

Description	Value	Units
Tangential cutting coefficient	2500	N/mm ²
Radial cutting coefficient	750	N/mm ²
Tool stiffness	5x10 ⁶	N/m
Damping ratio	0.05	-
Tool natural frequency	2.4	kHz

For this study, a Brownian distribution is used to describe the initial knowledge state (Table 2). In addition, it is assumed that the stability limit is known to occur below an axial depth g_{max} known *a priori*. While this may be an unrealistic assumption in practice, the bound provides an intuitive link between the imposed constraint and $c(x, g)$.

TABLE 2. Parameters defining the Brownian distribution prior to experimentation.

Description	Parameter	Value
Upper bound	g_{max}	1.2 mm
Diffusion field	$k(x, g)$	0.00005
Drift field	$\mu(x, g)$	0
Creation and killing field	$c(x, g)$	∞ for $g < 0$ ∞ for $g > g_{max}$ 0 otherwise

Here, new knowledge is gathered using only simple stability testing, where a machinist selects an axial depth-spindle speed combination, performs a test cut, and decides whether the cut was stable or unstable. For this numerical evaluation, the testing is simulated by comparing the selected test parameters $\{x_i^t, g_i^t\}$ to the predefined/reference stability limit.

Since simple stability testing updates knowledge of $g(x)$ directly, the probability density functional for any updated knowledge state is again a Brownian distribution, and can be expressed by "killing" all $g(x)$ which are not consistent with the experimental data. For an arbitrary number of simple stability tests, the updated creation and killing field is given by Eq. (3).

$$c_{up}(x, g) = \begin{cases} \infty & \text{for } g < 0 \\ \infty & \text{for } g > g_{max} \\ \infty & \text{for } g(x_i^t) < g_i^t \text{ if test } i \text{ stable} \\ \infty & \text{for } g(x_i^t) > g_i^t \text{ if test } i \text{ unstable} \\ 0 & \text{otherwise} \end{cases} \quad (3)$$

In order to gauge how performing an experiment will affect profit, the decisions that are affected by knowledge of the stability limit should be considered. In this study, the decision situation consists of selecting an axial depth-spindle speed combination $\{x_{op}, g_{op}\}$ at which to mill away a cube of material. Profit for stable operation, $P(x_{op}, g_{op})$, is calculated using the approach and parameters described in [3]. Profit for unstable operating parameters is assumed to be zero.

For an arbitrary test history, defined in Eq. (4) where R_i^t denotes the result of stability test i , the expected profit of a production run, $P_{exp}(x_{op}, g_{op})$, can be calculated using the probability tree shown in Fig. 2.

$$T_i := \{x_i^t, g_i^t, R_i^t\} \quad 1 < i < \# \text{ of tests} \quad (4)$$

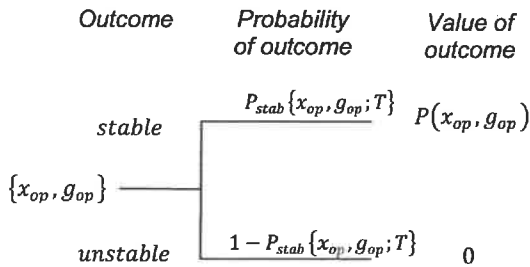


FIGURE 2. Probability tree for calculating the expected profit of a production run $P_{exp}(x_{op}, g_{op}; T)$.

For a stability test at an arbitrary $\{x^t, g^t\}$, the probability of each outcome, as well as the expected value of each outcome (stable or unstable) can be calculated before actually knowing the outcome of the test. Using the probability tree shown in Fig. 3, the expected value of the production run after the experiment's result may be calculated. Using larger probability trees, this calculation can be extended to an arbitrary number of tests, which can be optimized to determine an optimal testing policy.

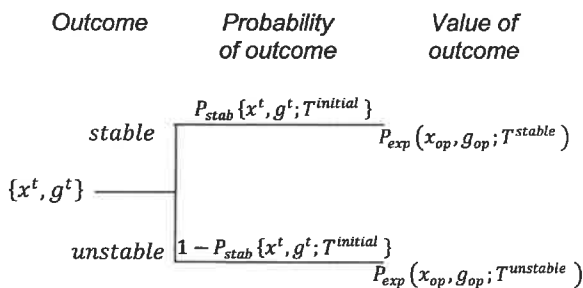


FIGURE 3. Probability tree for calculating the value of performing an experiment, $V_E\{x^t, g^t\}$. $T^{initial}$ is the current knowledge state, T^{stable} denotes $T^{initial}$ augmented with an additional row vector $\{x^t, g^t, stable\}$, and $T^{unstable}$ denotes $T^{initial}$ augmented with an additional row vector $\{x^t, g^t, unstable\}$.

Since stability tests can be chosen sequentially, the number of test parameters grows exponentially with the number of tests being considered. To reduce the computational complexity of this optimization, each test was optimized individually, instead of optimizing the full sequence of tests. Using this heuristic, a sequence of twelve experiments were chosen and tested against the reference stability lobe diagram. Figure 4 shows the progression of the predicted stability limit as the results of a sequence of experiments are incorporated into the prediction.

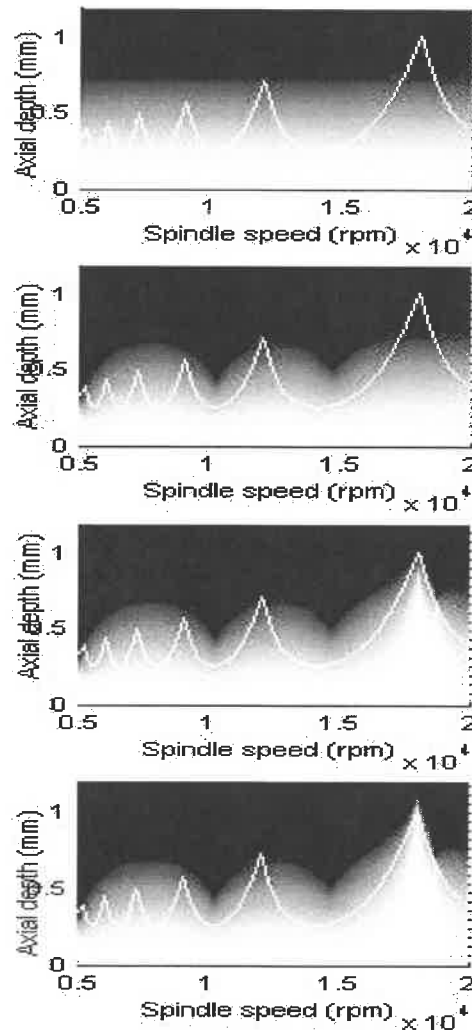


FIGURE 4. Progression of $P_{stab}\{x, g; T\}$ as experimental data is gathered. The top panel shows the prediction prior to any testing. From top to bottom, the remaining panels show the updated predictions after 4, 8, and 12 tests, respectively.

BUILDING GENERAL DISTRIBUTIONS BY TRANSFORMING A SET OF OBSERVABLES

While Brownian distributions offer the advantage of simplicity, in some cases it is advantageous to describe the function using an underlying model. For example, the stability limit can be inferred from knowledge of the frequency response function (FRF) and force model coefficients [4], which can be used as the set of measurables, instead of assigning a distribution over the stability limit directly. The advantage of this treatment is two-fold. First, a complex correlative structure is implied directly from the underlying dynamical model. Second, experiments which measure these parameters (such as force measurement or impact testing) can be incorporated into optimal experimental design. This provides an important alternative to simple stability testing, since a single FRF measurement enables much more accurate predictions to be made (see Fig. 5).

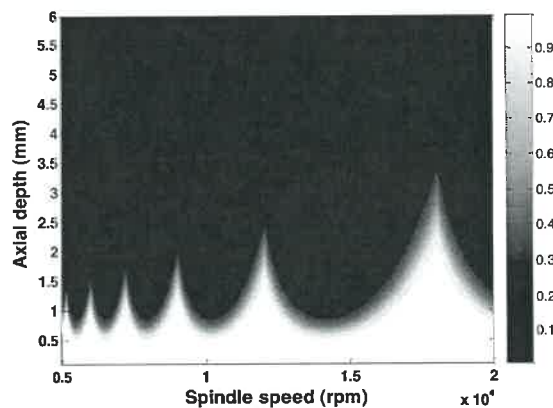


FIGURE 5. Contour plot of the probability of stability calculated assuming uncertain force model coefficients [3].

While the Brownian correlative structure works best when little is known about the function of interest, a correlative structure built from an underlying model enables predictions to be made when relationships governing the underlying dynamics are known.

CONCLUSION

As efficiency and productivity demands on the manufacturing industry increase, the need for a systematic approach to information gathering is growing. Due to the inherent complexity involved in many manufacturing decision problems, a general treatment of optimized experimentation for an arbitrary pay-off and knowledge state is

beyond the scope of traditional statistical methods. However, Bayesian methods are a natural candidate for this task. Not only can they treat arbitrarily complex mathematical objects, but they also associate a value with each knowledge state. As a result, information gathering can be optimized rigorously.

Since Bayesian methods have not previously been widely applied in manufacturing decision making, the primary motivation for this work was to explore possibility of using Bayesian methods to predict functions arising from the complex dynamics involved in milling. First, Brownian distributions were used to explore how Bayesian methods could be used in cases where little is known about the function of interest. Second, the process of incorporating underlying dynamical models into Bayesian prediction was explored. While these methods are still in their infancy, they offer sufficient analysis capability for a systematic treatment of the prediction of complex milling dynamics.

ACKNOWLEDGEMENTS

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FRICION MODELING OF LINEAR ROLLING ELEMENT BEARINGS IN HIGH PRECISION LINEAR STAGES

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INTRODUCTION

The goal of this effort is to develop an effective method of measuring the force of friction in linear rolling element bearings in a state that mimics conditions encountered in high precision stages. With useful data collected, three common complex friction models, the Dahl model [1], the LuGre model [2], and Generalized Maxwell-Slip [3] (GMS) model have been evaluated in terms of their utility in describing the system data. Since the focus of this effort is dealing with friction on industrial mechanical components in terms of its phenomenological effects, it seems reasonable to review previous studies geared toward the same situations.

The most common method for estimating the force of friction in these terms relies on scaling the current in this system's servo motor. Assuming that the time constant of the motor is sufficiently short and the motor constant is approximately ideal, the equation of motion (EOM) for such a system can be written as

$$m\ddot{x} + f_r = K_e i \quad (1)$$

where m is the moving mass, $\ddot{x}$ is the acceleration of the mass, f_r is the force of friction, K_e is the motor force constant and i is the current in the motor. Equation 1 can be rearranged to solve for the force of friction, this yields two quantities which must be solved for by using measured data. In most cases the assumption that servo feedback current scales linearly to the force applied to the system is valid. An accelerometer can be used to measure acceleration directly. However, the accelerations may be quite small and difficult to accurately measure even with a high quality accelerometer. An acceleration estimate could also be made by twice differentiating a position signal, but this approach amplifies quantization noise.

An alternate approach is proposed as a solution to some of the metrology limitations. A realistic testing condition is constructed if the rolling element bearings are installed in a linear stage and this structure is shaken axially. Such a configuration is shown in Figure 1, where a linear stage with rolling element bearings is provided a sinusoidal base excitation with a linear air bearing stage. The EOM for the upper moving mass, which is supported by the bearing to be studied, is

$$m\ddot{x} = f_r. \quad (2)$$

Thus, the force of friction can be simplified to a single quantity. The only mechanism for the transmission of force to the upper moving mass is the force of friction. It is important to note that $\ddot{x}$ in Equation 2 is acceleration in the inertial reference frame and not acceleration relative to the lower moving mass. In summary, this testing concept consists of an upper mass attached to a lower mass by a set of linear bearings. An accelerometer is used to measure the acceleration of the upper mass as the lower mass is actuated.

FRICION MODELS

The Coulomb friction model is not adequate to describe the force of friction on extremely small scales. On these length scales, the force of friction is better described as a non-linear stiffness that exhibits hysteretic properties. To approximate this behavior various models have been developed to describe this phenomenon.

The simplest and oldest of the more complex friction models is the Dahl model [1], defined as,

$$\frac{dF}{dx} = \sigma \left| 1 - \frac{F}{F_C} \right|^i \operatorname{sgn} \left(1 - \frac{F}{F_C} \right) \quad (3)$$

where F is the force of friction, x is generalized coordinate defining the axis of motion, σ is the initial contact stiffness, F_C is the approximate level of Coulomb sliding friction, and i is a shape factor parameter. This model is rate independent and does not account for possible Stribeck effect but, it does produce hysteretic behavior.

The LuGre model was developed to include possible rate dependence and account for possible Stribeck effect [2]. The LuGre model was constructed with the thought that surfaces subject to a frictional contact may be modeled as numerous contacting bristles which have individual stiffness and damping parameters. By averaging numerous individual interactions, the bulk phenomenon can be described by a simple model. The LuGre model is stated as,

$$\frac{dz}{dt} = v - \sigma_0 \frac{|v|}{s(v)} z \quad (4)$$

$$s(v) = F_C + (F_s - F_C) \exp \left(- \left| \frac{v}{V_s} \right|^\delta \right) \quad (5)$$

$$F = \sigma_0 z + \sigma_1 \frac{dz}{dt} + \sigma_2 v \quad (6)$$

where z is a hidden state describing the deflection of the bristles, v is the relative velocity between the surfaces, and the function $s(v)$ accounts for the Stribeck effect. The model of the Stribeck effect includes the level of kinetic Coulomb friction, F_C , the level of static friction, F_s , the Stribeck velocity, V_s , and a shape factor, δ , which is typically set to 1. The additional frictional constants are the micro-stiffness (bristle stiffness), σ_0 , the micro-damping (bristle damping), σ_1 , and the coefficient of viscous drag σ_2 .

One key shortcoming in both the Dahl and LuGre models is the lack of tunable parameters that are able to accommodate for arbitrary force versus displacement curves of frictional contacts. The Leuven model or Generalized Maxwell-Slip (GMS) model was developed to improve upon this [3]. The chief contribution of the GMS model is the incorporation of a hysteresis function instead of the simple hidden state, z , used in the LuGre model. This change effects the presliding characteristics of the GMS model while the gross sliding

characteristics are identical to the LuGre model. The GMS model is stated as

$$\frac{dz}{dt} = v \left(1 - \operatorname{sgn} \left(\frac{F_h(z)}{s(v)} \right) \left| \frac{F_h(z)}{s(v)} \right|^n \right) \quad (7)$$

$$s(v) = F_C + (F_s - F_C) \exp \left(- \left| \frac{v}{V_s} \right|^\delta \right) \quad (8)$$

$$F_f = F_h(z) + \sigma_1 \frac{dz}{dt} + \sigma_2 v. \quad (9)$$

Most of the parameters of the GMS model have the same interpretation as the LuGre model with the addition of the shape factor n and the hysteresis function $F_h(z)$. A Maxwell slip model is used to construct the hysteresis function. This model consists of a number of parallel saturating stiffness elements. In a single Maxwell slip element, the input is the hidden state, z , and the output is a force, which is one component of the overall force of friction. As the magnitude of the input z , increases, the magnitude of the output increases until a threshold is reached. At this point the element slips and force output will remain constant for any further increase in input magnitude. If at any point in time the rate of input, $\frac{dz}{dt}$, changes sign, stiffness behavior will resume until either the positive or negative threshold values are reached. Since the hysteresis function is likely to contain multiple Maxwell slip elements, the value of the hysteresis function can be written as

$$F_h(z) = \sum_{i=1}^N F_i \quad (10)$$

where N is the total number of Maxwell slip elements and F_i is the force output of each respective element.

STATIC ERROR

The primary component of the testing apparatus is the Aerotech ALS130H-100. This linear stage has an encoder resolution of 0.066 nm and a moving mass of 1.3 kg. The force constant for the actuator is listed as 7.93 N/A. The power amplifier is capable of providing ± 5 A with 16-bit digital resolution thus, the smallest increment of commandable current is 0.153 mA. However, there

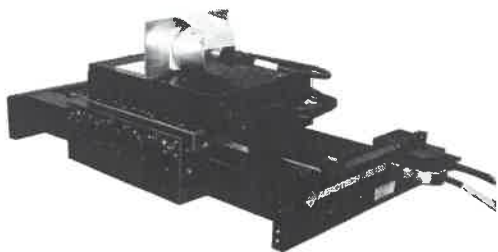


FIGURE 1. Testing Setup (bottom ABL1500, top ALS130H, PCB 393B31 accelerometer)

is a 0.630 mA_{rms} measurement error of the feedback current. The base motion is provided by an Aerotech ABL1500 airbearing stage. A picture of the test set up is shown in Figure 1

With knowledge of the parameters of the components in the test system, it is now possible to make estimates of static measurement error that will occur with the different testing approaches. The first case presented, the current scaling method, is that of Equation 1, when the inertial response is assumed to be negligible, $m\ddot{x} \approx 0$. This yields a friction measurement with a resolution of 1.21 mN subject to measurement noise of 5.00 mN_{rms} . If an accelerometer, such as the PCB 393B31, is added to the friction measurement apparatus noise from the accelerometer must also be considered. The accelerometer noise is approximated as $50 \mu\text{m/s}^2_{rms}$ white noise. With the addition of the necessary accelerometer mounting hardware, the moving mass of the system increases to 2.135 kg . This leads to an increase in measurement noise of 0.107 mN_{rms} , which, is a small increase in noise compared to the noise already induced by scaling force from motor current. In the base excitation approach, only an accelerometer measurement is needed to estimate the force of friction. Thus the predicted static error is the product of the increased moving mass and the measurement noise from the accelerometer or 0.107 mN_{rms} .

SNR ESTIMATES OF PROPOSED METHOD

It has been shown that the base excitation approach should introduce less static measurement error, however, it is also important to estimate the SNR under a realistic test condition to assure that the measurements will be useful. It is estimated that the force of friction in linear bearings will probably not exceed 0.5 N . Since relative motion of the bearing rails in the upper stage is desired to be rather small, initial analysis will

be done assuming that the upper mass is rigidly fixed to the base. Given the initial estimate regarding the force of friction, a relevant value for magnitude of the force of friction that likely would be located in the presliding regime may be 0.05 N . Applying equation 2 and dividing through the mass of 2.135 kg , estimates a base acceleration magnitude of 0.0234 m/s^2 . For the acceleration measurement, which scale directly to the force of friction, this yields an SNR of 50 dB , a value which would suggest the potential for making good measurements.

TESTING PROCEDURE

A model developed on both the larger displacement data and the smaller displacement data would best describe the observed phenomenon. Three test frequencies, 1.6 , 4 , and 8 Hz , have been selected to have a high amplitude relative displacement (tens of μm) and a low amplitude displacement (tens of nm). Since amplitude of the actual base excitation does not factor into the actual computation of the force of friction, this amplitude was adjusted until the desired relative displacement of the upper stage was near the desired amplitude.

The data used in the model evaluation are 5000 point time series sampled at 1 kHz . The metric of goodness of model fit to data is mean square error (MSE). To give a more equal weight to each of the respective objectives, the MSE of the high and lower amplitude model fits is multiplied by the inverse of the rms of the measured friction force data. In doing so, a given fitting error on the small amplitude data set is penalized more than the same magnitude error on the larger amplitude data set. The models used are those described in the previous section, the Dahl model, the LuGre Model, and the GMS Model. The GMS model used includes 8 Maxwell slip elements.

RESULTS

The results of these model fits are presented in Tables 1 and 2. As seen in Tables 1 and 2, the Dahl and LuGre models perform rather comparably in all trials, while the GMS model outperforms both the Dahl and LuGre models in all test cases. This increased performance of the GMS model is particularly evident when looking at the low displacement data sets. For further insight into nature of the improved performance of the GMS model, force versus displacement measurements are compared to model outputs for the low displacement and high displacement data sets in

Figures 2 and 3 respectively. Data from the LuGre Model has been omitted from the figures, as it is not much different from the Dahl model and makes interpretation more difficult. In Figure 2 it is seen that the GMS model is able to provide a better fit to this oddly shaped transition curve than does the Dahl model. In Figure 3, due to the lack of tunable parameters in the Dahl model (and also the LuGre model) almost no fit to the measured data is possible. If the Dahl model was weighted to fit the data of the small displacement in Figure 3, this would come at the cost poor fitting to the larger displacement data. However, with the addition of more tunable parameters in the GMS friction model, one model is able to accommodate both data sets. Additionally, these rolling element bearings seem to have a much greater apparent stiffness over small displacements than over somewhat larger displacements.

	1.6 Hz	4 Hz	8 Hz
Dahl Model	1.957	2.132	0.7524
LuGre Model	2.321	1.567	1.4884
GMS Model	1.430	1.229	0.2459

TABLE 1. MSE (mN^2) for high amplitude model fits

	1.6 Hz	4 Hz	8 Hz
Dahl Model	2.930	2.070	1.733
LuGre Model	2.906	2.109	2.220
GMS Model	0.345	0.054	0.038

TABLE 2. MSE (mN^2) for low amplitude model fits

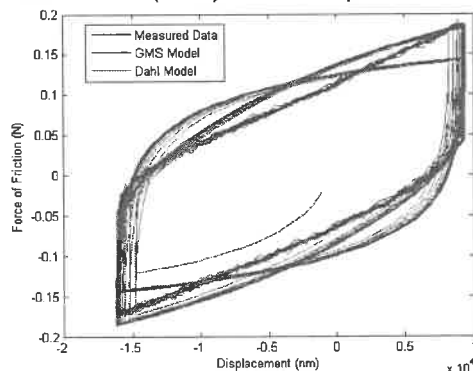


FIGURE 2. Comparison of measured data and models for large displacement ($\approx 25 \mu\text{m}$)

CONCLUSION

In this effort a different method of measuring the force of friction in linear rolling element bearings has been presented. While susceptible to some of the same sources of measurement error, the base excitation approach has been estimated to be far more precise in measuring the force of fric-

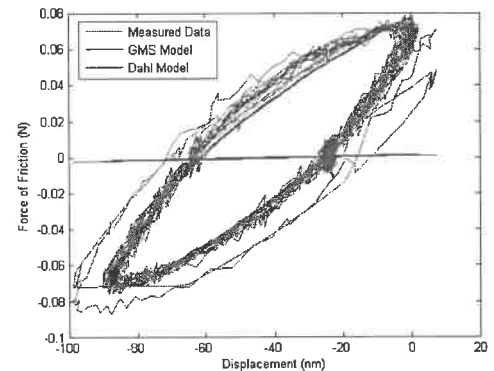


FIGURE 3. Comparison of measured data and models for small displacement ($\approx 100 \text{ nm}$)

tion than approaches which used scaled current to estimate force. Additionally, the quality of fit of three different friction models have been evaluated on data sets generated by a base excitation test fixture. It has been seen that GMS friction model, with its increased degrees of freedom, shows the best performance in modeling the force of friction over a wide range of displacements.

ACKNOWLEDGMENTS

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ACCURATE CONTROL OF BALL SCREW DRIVES USING POLE-PLACEMENT BASED VIBRATION DAMPING AND OPTIMIZED TRAJECTORY PRE-FILTERING

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INTRODUCTION

In order to meet industry demands for improved productivity and part quality, machine tools must be equipped with faster and more accurate feed drives. Over the past two decades, research has focused on the development of new control strategies and smooth trajectory generation techniques. These developments, along with advances in actuator and sensor technology, have greatly improved the motion delivery accuracy in high speed machine tools.

In ball screw drives, the high accelerations required for high speed motion may excite the structure's axial and torsional vibration modes. These modes place a limit on the achievable closed loop bandwidth, thus also limiting the positioning and trajectory tracking accuracy. Several strategies for dealing with these vibrations have been proposed in literature, including pre-filtering of the motion commands and notch filtering of the control signal to attenuate actuation near the resonances. The latter approach also allows a modest increase in the achievable control bandwidth, resulting in improved dynamic positioning accuracy. More recent techniques have focused on actively dampening the vibrations inside the control loop [1],[2]. This allows the bandwidth to be increased even further, but at the cost of a more challenging design, as the vibratory dynamics necessitate a higher order plant model to be considered.

In this work, pole-placement control is used to actively dampen out the 1st mode of axial vibrations in a ball screw. Trajectory tracking is improved through feedforward control, using inverted plant dynamics, feedforward friction compensation, and an optimally tuned trajectory pre-filter. Effectiveness of the proposed strategy is verified with impact hammer testing (for disturbance rejection around 1st axial mode) and trajectory tracking experiments. The proposed scheme is compared to the mainstream P-Pi cascade design used in industry.

CONTROLLER DESIGN

The controller is designed considering the simplified model shown in FIGURE 1, which captures the dynamics of the 1st axial mode [2] with sufficient closeness. Inertia of the rotating and translating components are denoted by m_1 and m_2 . k represents the overall axial stiffness, which is influenced by the nut, the fixed bearing, and torsional and axial stiffness of the screw. The damping in the preloaded nut is represented by c . Viscous damping at the motor and screw bearings, and table guide ways, is captured with b_1 and b_2 . The motor torque is represented in terms of the equivalent control signal u [V]. d_1 and d_2 correspond to equivalent disturbances acting on the rotating and translating components. The position units are in radians, which represent equivalent rotational motion of the screw. Further details about the setup can be found in [2].

This model was validated by comparing the

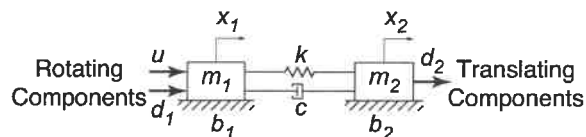


FIGURE 1. Simplified flexible drive model.

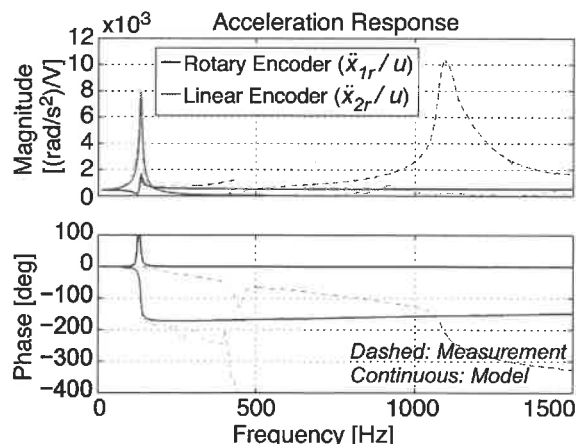


FIGURE 2. Measured and modeled drive FRF's.

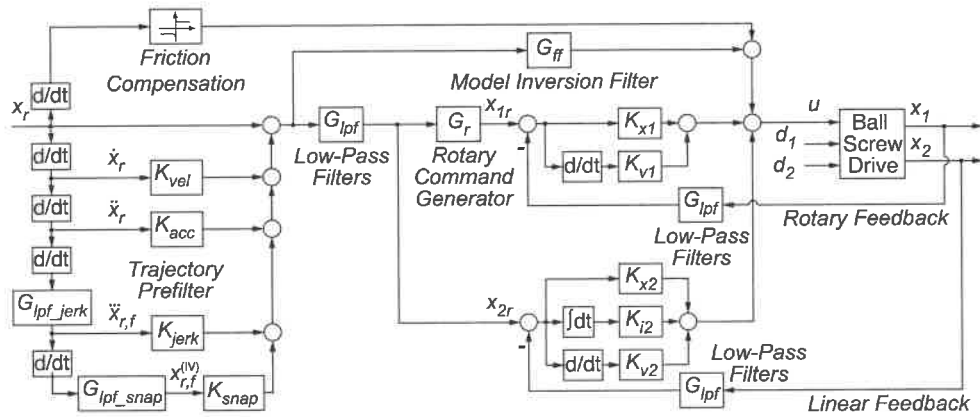


FIGURE 3. Proposed control scheme which compensates the 1st axial mode using pole-placement.

acceleration FRF measured through rotary and linear encoders, shown FIGURE 2. As can be seen, the 1st axial mode at 135 Hz is adequately captured. However, the gain and phase agreement deteriorates after ~250 Hz due to the additional dynamics contributed by torsional modes (450, 1160 Hz), and the bandwidth (450 Hz) of the drive's current control loop. The identified drive model comprises of complex conjugate poles ($s_{1,2} = -\sigma_1 \pm j\omega_{d1}$) at 135 Hz with 3% damping, a real pole at 0.07 Hz due to the interaction of viscous friction and inertia, and an integrator for velocity to position conversion.

The states to be controlled are the position and velocity of the rotating and translating components, which are practically measurable. It is also desired to eliminate the steady-state positioning errors on the nut side due to unmodeled disturbances and dynamic effects, like the motion loss in the preloaded nut [3] which cannot be captured with the linear model in FIGURE 1. Thus, the integrated table position $x_{2i} = \int x_2(t)dt$ is also considered as an extra state to be controlled, thereby resulting in the state vector $z = [x_1 \ x_2 \ \dot{x}_1 \ \dot{x}_2 \ x_{2i}]^T$. Hence the drive can be represented in state-space (Eq. (1)) and transfer matrix (Eq. (2)) form as:

$$\dot{z} = Az + [B_1 \ B_2 \ B_3] \begin{bmatrix} u \\ d_1 \\ d_2 \end{bmatrix}, \quad \begin{bmatrix} x_1 \\ x_2 \end{bmatrix} = Cz \quad (1)$$

$$\begin{bmatrix} x_1 \\ x_2 \end{bmatrix} = \begin{bmatrix} G_{11} & G_{12} & G_{13} \\ G_{21} & G_{22} & G_{23} \end{bmatrix} \begin{bmatrix} u \\ d_1 \\ d_2 \end{bmatrix} \quad (2)$$

The feedback controller is designed principally through the use of pole-placement control (PPC). The objective is to increase the dynamic

stiffness of the ball screw around the 1st axial mode, while also realizing a sufficiently high tracking and disturbance rejection bandwidth for rigid body dynamics. The model has 5 states, hence 5 pole locations are needed. Two of the poles ($p_{1,2}$) are placed to speed up the decay of axial vibrations without altering the oscillation frequency. A frequency change would be more costly in terms of control effort, since these poles originate from the drive's structure. Hence, only the real component $-\sigma_1$ is shifted further to the left in the s-plane, as: $p_{1,2} = -\beta\sigma_1 \pm j\omega_{d1}$, where $\beta \geq 1$. In stability analyses, it was seen that β could be increased up to 4.

The remaining three poles, which relate to the rigid body motion and integration effect in the servo controller, are placed at a frequency of 50 Hz. Two of them ($p_{3,4}$) are specified as complex conjugate with $\zeta_{3,4} = 0.35$. The fifth (p_5) is real. The state feedback gain $K = [K_{x1} \ K_{x2} \ K_{v1} \ K_{v2} \ K_{i2}]$ is computed to achieve $\text{eig}(A - B_1K) = \{p_1 \dots p_5\}$. Although $\zeta_{3,4} = 0.35$ may seem relatively low, most well-tuned servo controllers have a phase margin of 30°, which corresponds to complex conjugate poles with $\zeta = 0.3$. This allows the bandwidth to be increased as much as possible, without the control law becoming overly aggressive in the high frequency range. The implementation of PPC, as a PD and a PID controller, can be seen in the overall scheme shown in FIGURE 3.

In practical implementation, additional dynamics due to the torsional modes and motor current loop cause noticeable discrepancies between the model and the actual drive after 250 Hz.

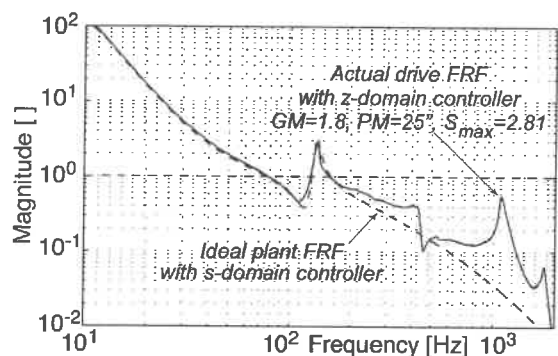


FIGURE 4. Loop magnitude of feedback design.

These can result in serious stability problems unless their effect is attenuated. Hence, additional low-pass filters are inserted into the feedback and feedforward channels, to treat the commands and measurements in a consistent manner, as shown with G_{lpf} in FIGURE 3.

These filters comprise of two real poles, one at 500 and the other at 1100 Hz.

The stability of the design and tuning of the pole locations and low-pass filters were realized using frequency domain analyses. Although the system has two outputs, the fact that there is only one controlled input allows SISO-type frequency domain analyses to be carried out, by considering the loop transfer function $L(s)$:

$$L = K_1 G_{11} + K_2 G_{21}, \quad \begin{matrix} K_1 = K_{x1} + K_{v1}s \\ K_2 = K_{x2} + K_{i2}/s + K_{v2}s \end{matrix} \quad (3)$$

Above, s is the Laplace operator. G_{11} and G_{21} were defined in Equation 2. For $\beta=4$, the magnitude and Nyquist plots of the loop transfer function are shown in FIGURE 4 and FIGURE 5. The figures show both the idealized case, with the theoretical model coupled to the ideal “s-domain” controller; and the actual implementation obtained by combining the experimental drive FRF together with the discretized version of the control law. The latter also includes the effects of the ZOH and a full sample (i.e. worst-case) computational delay. The sampling frequency was 16 kHz. Derivative and integration operations were implemented using Euler approximation. The low-pass filters were discretized using pole-zero matching.

Considering FIGURE 4 and FIGURE 5, the system has the desirable characteristics of loop shaping design: low frequency gain amplification and high frequency attenuation. The ideal and

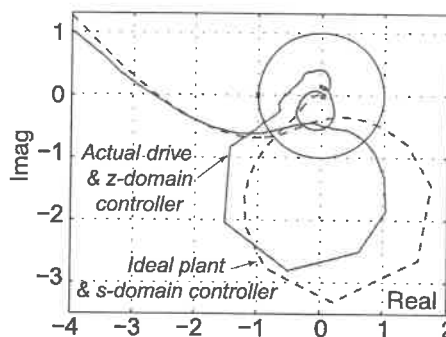


FIGURE 5. Nyquist plot for designed controller.

actual FRF's are in agreement up to 170 Hz. The first crossover occurs at 78 Hz, which is reasonably high for controlling the rigid body dynamics. The second magnitude increase is between 126-156 Hz, which corresponds to active damping of the 1st axial mode. The low-pass filters attenuate the torsional anti-resonance at 450 Hz and resonance at 1160 Hz, and prevent them from interfering with the loop stability. The stability margins, shown in FIGURE 4 are slightly lower than commonly used values, as the controller was tuned to favor higher tracking accuracy. If better robustness is desired, the rigid pole frequencies and attenuation factor β can be detuned.

Following the feedback design, it is essential to design the feedforward terms such that accurate command tracking can be realized. Since command x_r corresponds to the desired table motion (x_{2r}), the flexibility of the ball screw must be considered when generating the rotary motion commands. This is achieved by the filter $G_r(s)$ in FIGURE 3. Noting m_2 in FIGURE 1 and ignoring d_2 , $G_r(s)$ can be found as:

$$x_1(s) = \underbrace{\frac{m_2 s^2 + (b_2 + c)s + k}{cs + k}}_{G_r(s)} x_2(s) \quad (4)$$

Additional feedforward action is provided by inverting the plant dynamics using $G_{ff}(s)$:

$$G_{ff}(s) = 1/G_{21}(s) \quad (5)$$

G_{ff} and G_r are discretized using pole-zero matching and appending necessary delay terms to make these filters causal.

When the ideal plant and s-domain controllers are considered, the inclusion of G_r and G_{ff} enables a perfect tracking transfer function of

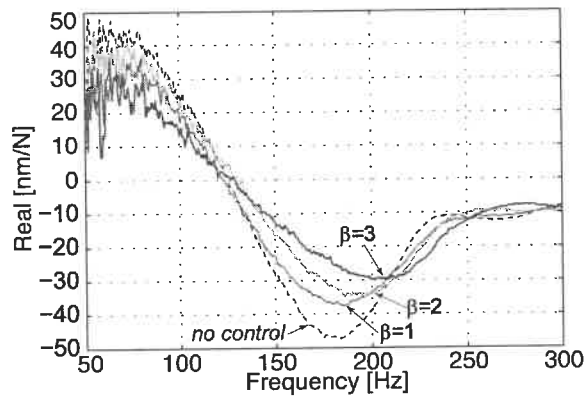


FIGURE 6. Real component of axial disturbance TF, measured through impact hammer testing.

unity. However, due to filter and computation delays, and variation of the actual drive FRF from the ideal model, additional adjustment needs to be made to the position commands. Otherwise, artifacts correlated to the velocity, acceleration, jerk, and snap commands may appear in the tracking error profile. To address this problem, a trajectory pre-filter is added, as shown in FIGURE 3. The filter gains K_{vel} , K_{acc} , K_{jerk} , K_{snap} are computed using Least Squares technique to allow the prediction of the observed translational tracking error ($e_2 = x_r - x_2$) as closely as possible, as a function of the commanded kinematic profiles:

$$\hat{e}_2 = K_{vel} \dot{x}_r + K_{acc} \ddot{x}_r + K_{jerk} \dddot{x}_{r,f} + K_{snap} x_{r,f}^{(IV)} \quad (6)$$

A single tracking experiment, with the gains set to zero, is sufficient for identifying the filter coefficients. In order to avoid numerical problems with high order derivatives, 2nd and 8th order low pass filters are used with 500 Hz pole frequency, shown with G_{lpf_jerk} and G_{lpf_snap} .

EXPERIMENTAL RESULTS

The dynamic stiffness enhancement achieved through the proposed design was measured using impact hammer testing on the table. Considering FIGURE 6, when $\beta = 3$, the minimum real value of the disturbance transfer function is reduced from -47 nm/N at 182 Hz (no control case), to -29 nm/N at 205 Hz, indicating a 38% improvement. This helps to improve the drive's immunity against machining chatter vibrations [4].

The proposed scheme was also compared to the industry standard P-PI translational position

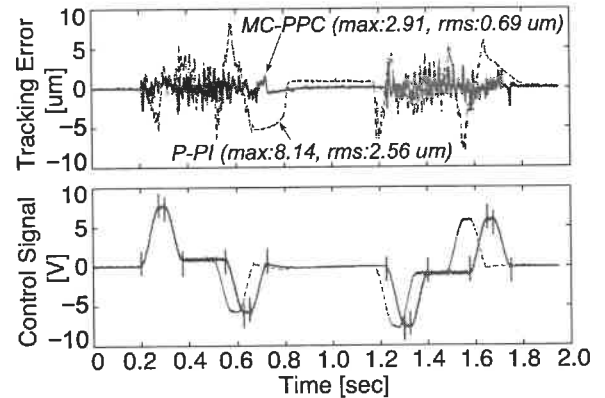


FIGURE 7. Tracking comparison between P-PI cascade and mode-compensating controller.

and rotational velocity controller. The P-PI implementation included notch filters at axial and torsional modes, inertia and damping feedforward terms, and Striebeck-type friction pre-compensation [2]. This controller was tuned to yield the best possible tracking accuracy, while retaining stability margins comparable to the proposed design. The velocity and position loop crossover frequencies were 152 and 27 Hz. 300 and 350 mm jerk-continuous motion was commanded with 1 m/s velocity, 10 m/s² acceleration, and 200 m/s³ jerk. The resulting profiles are shown in FIGURE 7. As can be seen, the proposed controller (MC-PPC) significantly outperforms the P-PI design.

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APPLICATION OF AN ACTIVE CLAMPING SYSTEM IN WOOD MACHINING

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INTRODUCTION

The productivity demands put on wood-working machines are continuously increasing. Generally, this resulting increase in productivity also leads to greater dynamic and in-process forces, which may cause higher structural vibration amplitudes during machining and thus negatively affect the production quality [1]. Therefore, technological advancements are made to improve the performance and machining speeds of these machines.

Vacuum clamping systems are predominantly used on stationary wood-working machines during machining of particle boards and fiberboards. Using these systems, the workpieces are neither damaged nor deformed during milling, drilling or sawing processes. These clamping systems also allow an arbitrary positioning of each suction block and good accessibility to workpiece edges. However, they have the disadvantage that the workpieces have free, overhanging ends. These ends are perturbed during the machining over a wide frequency range, thereby inducing forced vibrations. Moreover, these vibrations increase the risk of flaws along the edges of the workpiece. The quality of the workpiece edges, however, is an important quality criterion for the final product.

Furthermore, Figure 1 shows that the forced vibrations through the workpiece during milling process bring about higher overall noise levels, both direct and indirect sounds, which lead to elevated noise exposure to the production environment. The direct sound radiates originally from the milling tool alone, while the indirect sound comes from the workpiece through impulse interaction of the milling tool.

Generally, local encapsulation and passive noise damping methods are used to counteract this problem, but they do not reduce the noise level as desired. Besides, particularly for the stationary wood-working machines, such solutions are constructively limited. Thus, an improved performance can only be realized through innovative fundamental system analysis. This kind of improvement may be achieved by the integration of active or adaptronic (a.k.a. smart structure) systems into the mechanical components of machine tools [2]. Therefore, the idea was to design and employ an active clamping system on stationary wood-working machines.

Thus, the goal of the research project presented in this paper is to achieve an active vibration and noise level reduction during machining of particle- and fiberboards on stationary wood-working machines.

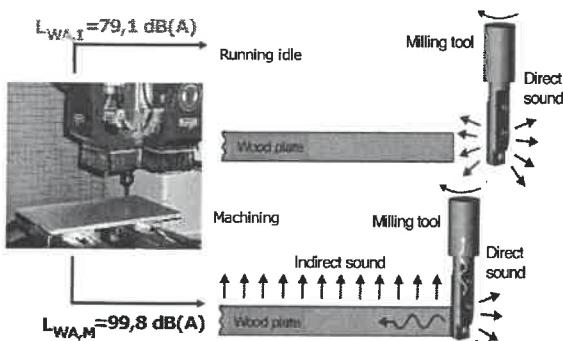


FIGURE 1: Noise exposure during milling process

MODELING AND DESIGN

Initially, an adaptronic clamping system, which is composed of sensors, a workpiece clamped on four suction blocks and actuators within each of those suction blocks, was modeled into a multi-body-simulation (MBS) environment (Figure 2). The simulation aims to analyze its fundamental behavior. The workpiece was modeled as a finite-element-model and imported into MSC.ADAMS over an interface. Subsequently, an excitation force and actuator's reactions applied to the workpiece were also modeled. The excitation force was implemented in the MBS-model as an input vector. This vector was modeled on a workpiece edge and defined

as a sine signal. The piezoelectric actuators were likewise modeled to react in the sine signals with the same amplitude, however in the different phase from the amplitude and phase from the input vector, respectively.

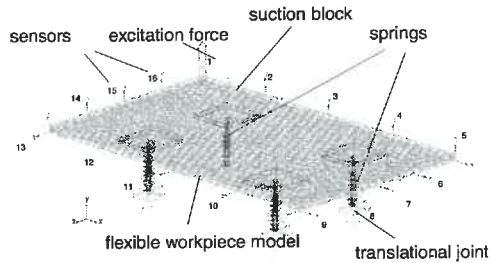


FIGURE 2: MBS-model of the adaptronical clamping system

In the following, the structure behavior was simulated in the MBS-environment. The simulation result in Figure 3 shows that the vibration amplitudes of the workpiece can be reduced about 14 times [3].

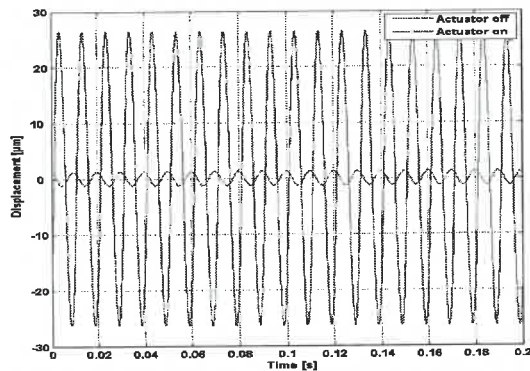


FIGURE 3: Time signal of the simulated system

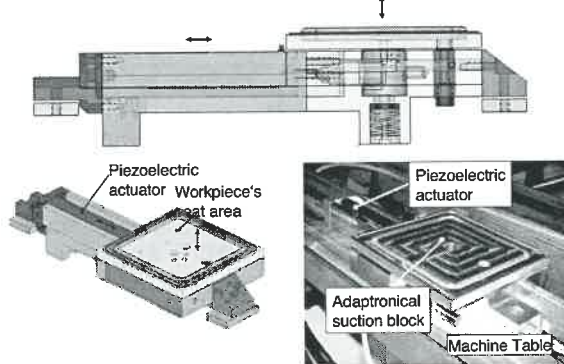


FIGURE 4: Adaptronical suction block

Based on the conducted simulation results, each adaptronical suction block was particularly designed under diverse actuator positions and possible actuating mechanisms in order to

obtain an ideal place on the clamping system. As shown in Figure 4, the piezoelectric actuator was located in horizontal position and its actuating movement is later on translated into a vertical one. It is carried out by a guided adapter with a sliding wedge interface inside.

CONTROL ALGORITHM

By using an active vibration control system, an external voltage signal is applied to the actuators to produce forces and accordingly displacements in the opposite direction of those produced by the tool at a particular workpiece position. The simple idea behind is the destructive wave interference and thus reduces vibrations. To achieve a wider amplitude vibration reduction area on the workpiece, several sensors and actuators are employed by implementing a Multi-Input Multi-Output (MIMO) system. Using a Filtered-X algorithm in the control strategy with adaptive filters Least Mean Square (LMS) and/or Recursive Least Square (RLS), the control system is intended to be adaptive in order to cope with various machining parameters and workpiece dimensions.

Each control signal from a single actuator, which propagates through the 'physical' system $S(z)$, is modified by the response characteristics of the control source and the workpiece between the control source and the error sensor. The influence of all of these can be lumped into a single 'secondary path transfer function' $S'(z)$, which is constructed through system identification [4].

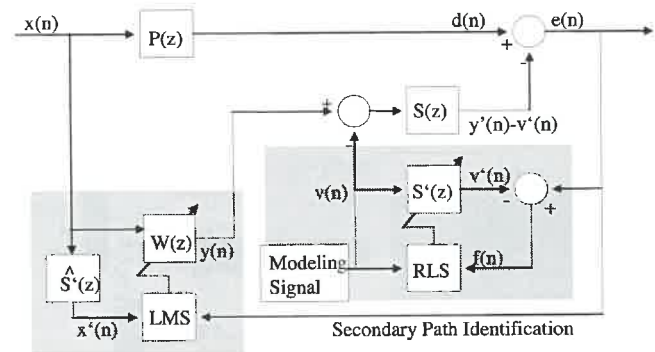


FIGURE 5: Block diagram of Filtered-X LMS algorithm using RLS cancellation path modeling

As illustrated in Figure 5, the error signal $e(n)$ measured by the error sensor is a mixture of signals both from $P(z)$ and $S(z)$. $P(z)$ is the primary path from the reference signal depending on the rotating frequency of the

milling tool to the error sensor, whereas $S(z)$ is the single secondary path between each piezoelectric actuator and each error sensor.

Due to the higher stability and less computation of the LMS, it is chosen to be used in the control process and the identification process is left to the RLS algorithm. Regarding the RLS performance, it is possible to identify the secondary path faster and more precisely. This combination leads to the Filtered-X LMS controller with the RLS identification, shortened to the FXLMS-RLS.

Assuming that $W(z)$ is a FIR- (finite impulse response-) filter of tap-weight length N , the control signal $y(n)$ is computed as

$$y(n) = \mathbf{w}^T(n) \mathbf{x}(n) \quad (1)$$

$$\mathbf{w} = [w_0(n), w_1(n), \dots, w_{N-1}(n)]^T \quad (2)$$

is a tap-weight vector and

$$\mathbf{x}(n) = [x(n), x(n-1), \dots, x(n-N+1)]^T \quad (3)$$

is an N -sample reference signal vector.

The tap-weights of the adaptive filter $W(z)$ are updated using the LMS algorithm [4], [5], [6].

$$\mathbf{w}(n+1) = \mathbf{w}(n) - \mu \mathbf{x}'(n) e(n) \quad (4)$$

where μ is the corresponding step-size parameter and $\mathbf{x}'(n)$ is an N -sample filtered reference signal. The filtered reference signal $\mathbf{x}'(n)$ is an approximation result after the influence factors of the cancellation path described as follows

$$\mathbf{x}'(n) = \hat{S}'(z) \mathbf{x}(n) \quad (5)$$

where $\hat{S}'(z)$ is copied from the cancellation path modeling $S'(z)$.

The tap-weights of the adaptive filter $S'(z)$ are updated using the RLS algorithm [6].

$$\mathbf{k}(n) = \frac{\lambda^{-1} \mathbf{P}(n-1) \mathbf{v}(n)}{1 + \lambda^{-1} \mathbf{v}^T(n) \mathbf{P}(n-1) \mathbf{v}(n)} \quad (6)$$

$$\mathbf{v}'(n) = \mathbf{s}^T(n) \mathbf{v}(n) \quad (7)$$

$$\mathbf{f}(n) = \mathbf{e}(n) - \mathbf{v}'(n) \quad (8)$$

$$\mathbf{s}(n) = \mathbf{s}(n-1) + \mathbf{k}^T(n) \mathbf{f}(n) \quad (9)$$

$$\mathbf{P}(n) = \lambda^{-1} \{ \mathbf{P}(n-1) - \mathbf{k}(n) \mathbf{v}^T(n) \mathbf{P}(n-1) \} \quad (10)$$

where $\mathbf{k}(n)$ is the Kalman gain vector, $\mathbf{s}(n)$ is the filter coefficients of $S'(z)$ and $\mathbf{P}(n)$ is the inverse of the auto-correlation matrix of $\mathbf{v}(n)$.

Since the designed control system is composed by four actuators and four error sensors, there are 16 secondary paths from each actuator as

secondary source to each error sensor (Figure 6).

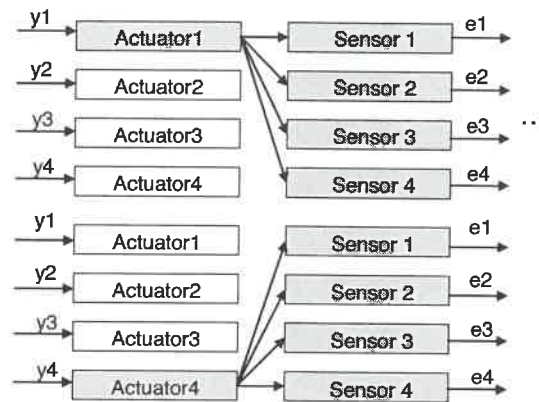


FIGURE 6: Multiple channel secondary path identification procedure

EXPERIMENTAL RESULTS

In order to examine the practical issues associated with the MIMO-FXLMS-RLS controller, several experiments during machining were done. Figure 7 shows the experimental setup with the MIMO-FXLMS-RLS structure using four LASER triangulation sensors (LTSs) as error sensors. The actuators and the sensors were located relatively close to the disturbance source at the milling position, yet the machining shall not be disturbed. The workpiece, a medium density fiberboard (MDF) in 600 x 600 x 18 mm³, was clamped on the active suction blocks. It was milled at a workpiece edge with a depth of cut of 2 mm and a tool's rotation speed of 18000 rpm.

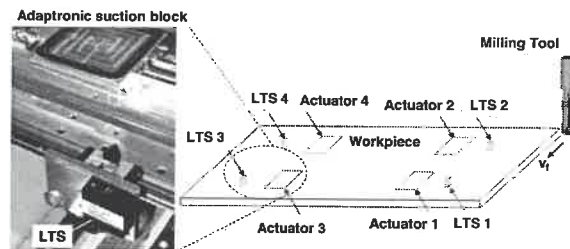


FIGURE 7: Experimental setup

During the milling process, the vibration amplitude of the workpiece was dominated at the tool's rotation frequency of 300 Hz. Therefore, the error signals were filtered by a narrow band-pass filter in the corresponding frequency. Each secondary path from each actuator to each error sensor was identified at the frequency of 300 Hz as well. The experimental result in Figure 8 shows an amplitude reduction in all error signals after the control was activated.

The filtered error signals taken from error sensors LTS 1, LTS 2 and LTS 3 -which were relative nearby to the disturbance source- had greater amplitudes as expected than one from error signal LTS 4. Since an improved workpiece quality is the primary research goal, the vibration amplitudes of the workpiece at the milling position (LTS 1 and LTS 2) become a priority to be reduced. A maximum vibration reduction was achieved by 25 dB. At the same time, the vibration amplitude at sensors LTS 3 and LTS 4 were reduced as well.

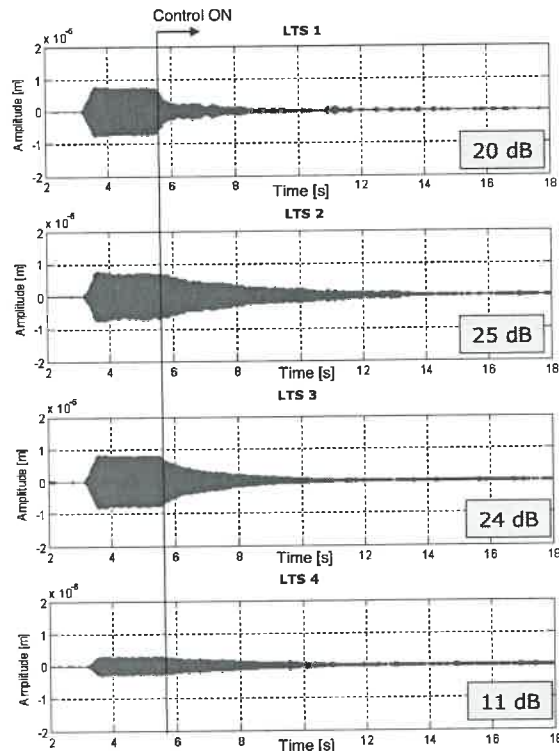


FIGURE 8: Error signals at milling process

Furthermore, in order to capture the noise radiation particularly from the workpiece during the machining, a microphone was subsequently positioned over the workpiece surface. The dominant sound pressure levels were measured likewise at the tool's rotation frequency of 300 Hz and its five harmonics at 600 Hz, 900 Hz, 1200 Hz, 1500 Hz and 1800 Hz.

By employing an accelerometer right in the middle of the workpiece surface as an error sensor, the active clamping system was also investigated in aim to reduce the noise levels at the tool's rotation frequency of 300 Hz and its three harmonics at 600 Hz, 900 Hz and 1200 Hz. Actuator 1 and 2 controlled the frequencies

of 300 Hz and 600 Hz, whereas actuator 3 and 4 handled the frequencies of 900 and 1200 Hz, respectively. The experimental result in Figure 9 shows an overall noise level reduction of 4 dB(A), in which a maximal noise level reduction at 300 Hz was achieved by 20 dB(A).

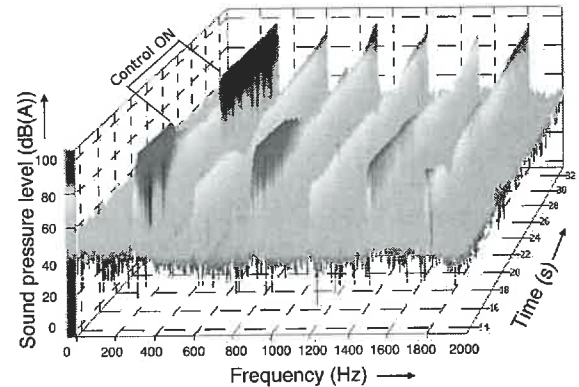


FIGURE 9: Noise levels at milling process

CONCLUSIONS

In this paper, it has been proved that with the developed adaptronic/active clamping system and the implemented MIMO-FXLMS-RLS controller the vibration amplitudes on the workpiece surface and the noise levels during machining are reduced by maximal 25 dB and 20 dB(A), respectively. The designed controller has provided a good convergence and a superior stability behavior.

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A STUDY ON DESIGN OF CONTROLLERS FOR SIX DEGREE OF FREEDOM ACTIVE VIBRATION ISOLATION SYSTEM

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INTRODUCTION

Nano precision instruments with nanometer resolution are susceptible to disturbance transmitted mostly from ground vibration and mechanical vibration due to movements of the measuring instrument itself. These kinds of vibrations which are submicrometer level and low frequency distract both precision and accuracy of precision instruments.

Six degree of freedom active vibration isolation system is required for these kinds of vibration. In this system, control algorithms are very important factors of performance improvement. The system of this sort has coupling and system uncertainties due to changing of environment.

The objective of this work is to design controllers which are PI controller and H-infinity controller for six degree of freedom active vibration isolation system and compare two controllers through experiments.

VIBRATION

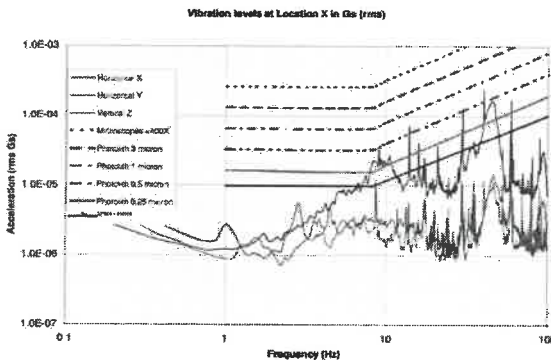


FIGURE 1. Ground vibration levels.

Ground vibration levels illustrate in Figure 1. Sub-micrometer level ground vibrations exert a bad effect to precision instruments. Figure 2 describes vibration criterion defined by ISO.

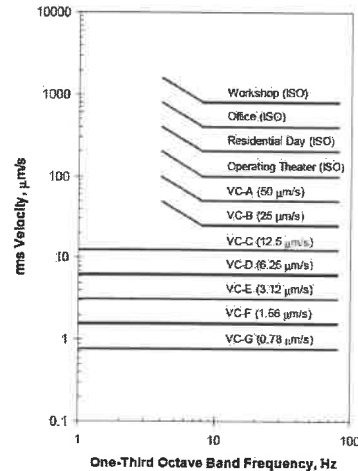


FIGURE 2. Generic vibration criterion (VC) curves for vibration-sensitive equipment.

Generally precision measuring and actuating system as AFM (Atomic force microscopy) and lithography instruments follow VC-E criterion. Recently, more tight criterions are required. Ground vibration isolation is very important issue to satisfy criterion.

OVERVIEW OF THE ACTIVE VIBRATION ISOLATION SYSTEM

In this paper, six degree of freedom active vibration isolation system composed by passive isolation to attenuate high frequency vibration and active isolation to attenuate low frequency vibration.

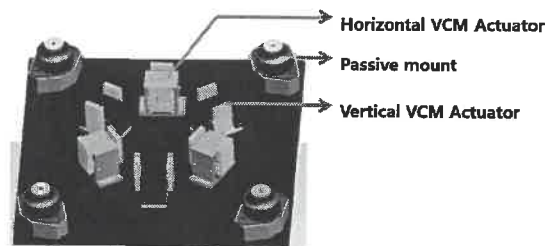


FIGURE 3. Six degree of freedom active vibration isolation system.

FIGURE 3 describes six degree of freedom active vibration isolation system used for experiment. This system uses four rubber mounts for supporting payload and passive isolation, six VCM (Voice Coil Motor) actuators for active isolation by actuating horizontal and vertical directions, and six accelerometers for sensing and feedback six directional acceleration signals of upper plate.

PASSIVE ISOLATION SYSTEM MODELING

Passive mount assumed to have stiffness and damping properties typically represented by spring in parallel to a damper.

FIGURE 4 shows conceptual diagram of passive isolation system both XY and XZ plane.

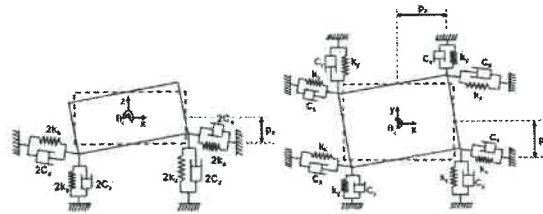


FIGURE 4. Conceptual diagram of passive isolation system.

Passive mount assumed to have stiffness and damping properties typically represented by spring in parallel to a damper. To effectively model the system consisting of viscoelastic material, some assumptions were made

© The principal axes of inertia of the payload are, respectively, parallel to the principal elastic axes of the mounts, so product moment of inertia $I_{xy} = I_{yz} = I_{zx} = 0$

© Consider the upper plate and the payload as one rigid body.

© The forces (F_x , F_y , and F_z) and moments (M_{θ_x} , M_{θ_y} , and M_{θ_z}) are applied directly to the upper plate.

© Ignore rotational stiffness and damping, because the motions of payload and lower plate make compressions and tensions much more than angular motion for passive mounts.

This system can be modeled as following equation.

$$([M]s^2 + [C]s + [K])P(s) = ([C]s + [K])B(s) + F_u(s) + F_d(s) \quad (1)$$

M, C and K are mass, damping and stiffness matrices, F_u is input force, and F_d is direct disturbance force respectively. P is six axis directional motion of the payload, and B is six axis directional motion of the ground vibration.

DESIGN OF CONTROLLERS

The purpose of active control is to effectively decouple the isolator frequency from the static sag requirement so that it can improve the low frequency isolation.

Two kinds of control method, PI control and H-infinity control, are proposed and designed to experiment on the six degree of freedom active vibration isolation system and compare performance of two controllers.

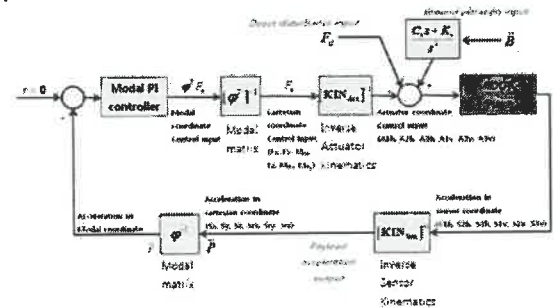


FIGURE 5. Modal decoupled PI feedback control loop.

FIGURE 5 shows block diagram of the modal decoupled closed loop acceleration feedback PI control. It has modal matrix for modal decoupling and kinematics of actuator and sensor which are obtained by their geometrical position. PI controller must be designed six decoupled mode and conform to required transmissibility for the system performance.

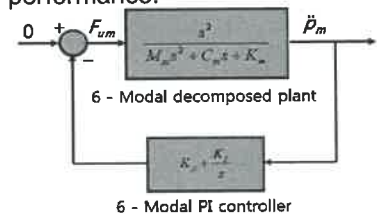


FIGURE 6. Block diagram of six modal PI controller

Relationship of each mode and PI controller can be simply represented as FIGURE 6. Using this relationship, equation (2) is derived and PI controller is can be designed using this equation.

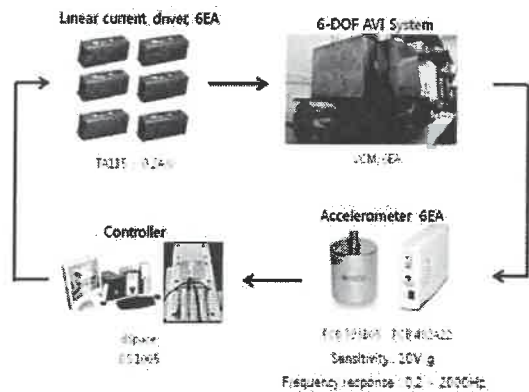
$$(M_m + K_p)s^3 + (C_m + K_I)s^2 + K_m s = 0 \quad (2)$$

FIGURE 7 shows block diagram of closed loop acceleration feedback H-infinity control. Δ , K_∞ and W are system uncertainty, H-infinity controller and weighting function respectively. Purpose of this control is to minimize the effect of z by the external input d . It can be represented as equation (3).

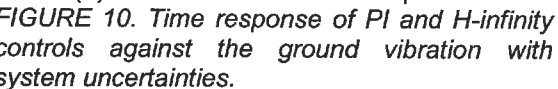
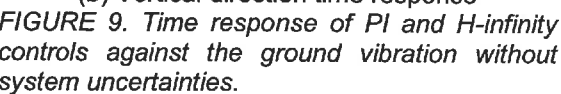
When S is

Two weighting functions must be defined properly. W1 is guideline of system sensitivity function which is determined by required transmissibility. W2 prevent saturation of actuator and is defined to ensure rank condition solving matrix equation of controller. Then minimum γ and controller K_∞ which are satisfied equation (3) can be got. H-infinity controller is obtained by using robust control toolbox of the MATLAB.

To verify the performance of two controllers, experimental system setup as illustrated in FIGURE 8 and two controllers are applied to the system.



To compare performance of ground vibration isolation and robustness of two controllers, experiment carried out with two conditions which are condition without system uncertainties and condition with system uncertainties. Time response and transmissibility are checked on each condition.



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payload as illustrated in FIGURE 11 and FIGURE 12.

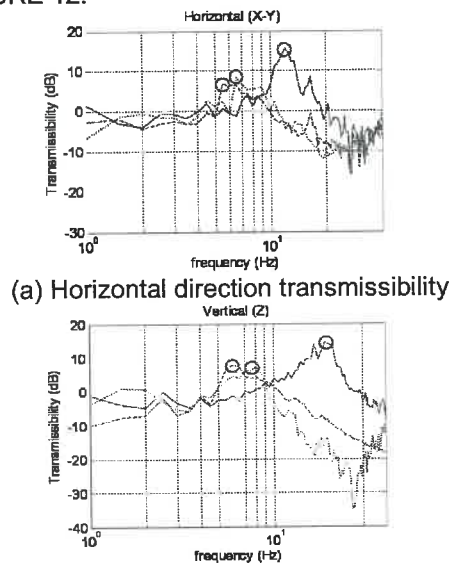


FIGURE 11. Comparison of transmissibility from the ground vibration to the payload between PI and H-infinity control without the system uncertainties. (solid – passive, dot – PI control, dash – H-infinity control)

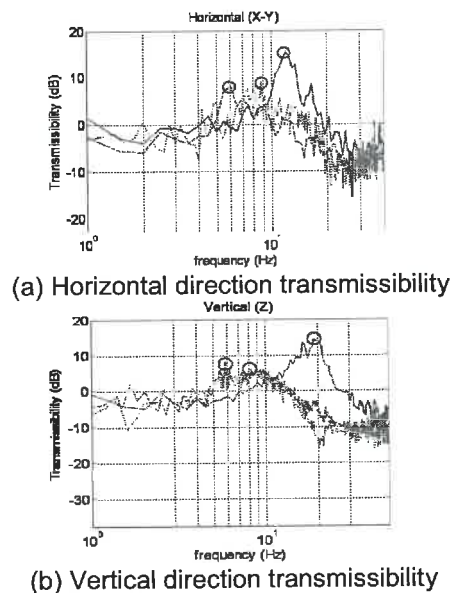


FIGURE 12. Comparison of transmissibility from the ground vibration to the payload between PI and H-infinity control without the system uncertainties. (solid – passive, dot – PI control, dash – H-infinity control)

These situations come from inaccuracy of modal decoupling of the PI control by the system uncertainties. The system uncertainties affect to

the system parameter values of inertia, damping and stiffness. On the other hand, H-infinity control based on robust control shows better performance with the system uncertainties.

CONCLUSION

In this work, modeling of the six degree of freedom active vibration isolation system was carried out. For system control, PI controller which is easy to design and H-infinity controller which is robust to uncertainties and disturbances are designed and proposed to control active isolator and applied to six degree of freedom active vibration isolation system. And then, performance of the PI controller is compared with the H-infinity controller.

In conclusion, the H-infinity controller is more compatible with six degree of freedom active vibration isolation system which has coupled axis and the various system uncertainties.

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TRACKING CONTROL OF THE BALL BEARING GUIDED LINEAR STAGE IN HIGH VACUUM

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INTRODUCTION

Recently, there are increasing needs for precision stages with vacuum compatibility in such as lithography equipment for wafer or mask manufacturing, mask mastering equipment for optical data storage and scanning electron beam inspection for semiconductor process. The precision stage in vacuum can have several types of bearing: vacuum compatible air bearing, magnetic bearing and vacuum compatible ball bearing. The vacuum compatible air bearing and magnetic bearing have frictionless motion and a few advantages such as smooth motion control result and controllable parasitic direction stiffness. However, these bearings require a complex bearing component design and a control method and an unstable motion can occur when abrupt motion is applied to air bearing stage due to degraded vacuum-ambient equilibrium. A ball bearing is used in high vacuum with proper material selection and surface finish or treatment. A simpler design, a high stiffness and a lower cost can be achieved by adopting ball bearing in high vacuum ($\sim 10^{-5}$ Torr). The ball bearing in vacuum also has friction and wear problems as in ambient air and when tracking control is performed, friction can make the tracking error higher and there may be a cold-weld phenomenon between ball and rail surface in high vacuum. Therefore, an adequate control method should be used for tracking control of the ball bearing stage especially in high vacuum. In case of higher speed tracking, tracking error will be increased due to fast desired path and in case of lower speed control, there will be control problems due to frictional forces between ball and guide surfaces. Therefore, a controller which is robust to parameter variation and which has fast tracking capability is needed.

The feedforward controller, ZPETC (Zero Phase Error Tracking Controller), will be adopted for

minimizing tracking phase error. The ZPETC control method implements pole-zero cancellation and zero phase tracking error is achieved theoretically [1]. However, the servo loop should have enough robustness for ZPETC to operate properly and therefore TDC (Time Delay Control) control method will be adopted simultaneously (exactly, in series with ZPETC). The TDC can make a stage follow a predetermined reference-model although there are uncertainties in model parameters and unpredictable external disturbances [2]. A pre-study with conventional PID controller was performed and in that case the control system showed somewhat poor performance which may be by friction existence of the ball bearing. In order to decrease friction, special surface finish, DLC (Diamond Like Carbon) film coating, is to be done on the surfaces of bearing component by sputtering deposition for vacuum compatibility.

Figure 1 shows the picture of the linear stage in a vacuum chamber. The linear stage was designed with linear motor and ball bearing guide (THK HR series). The linear motor is a

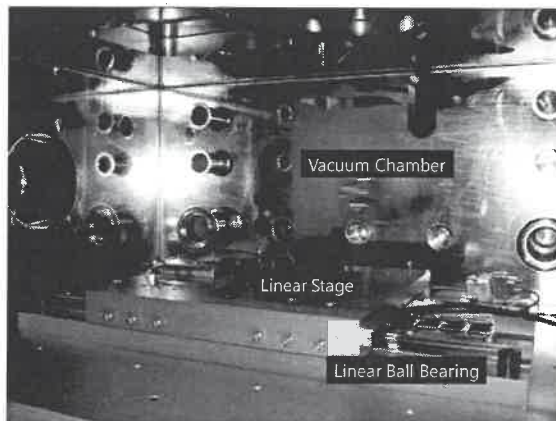


FIGURE 1. Picture of the linear stage. Linear motor, linear encoder and roller guides are shown.

coreless type motor and the linear position is measured by linear encoder (Heidenhain LIP481) and the encoder can have 1.25 nm resolution and the moving range of stage is 250 mm. Generally special lubricating grease, which has very low vapor pressure, is used in vacuum. However, this vacuum compatible grease generally makes bearing setup difficult and cause more friction on the stage movement. When the movement speed of the linear stage is below about 1 mm/s, it is too slow for the stage to have heat generation and wear problem. The surface of the bearing and guide is cleaned by solvent solution in order to remove any organic grease or inorganic dirt. And therefore, the surface contact between guide and bearing is a kind of dry-contact type. There was a research report regarding dry condition metal contact (that is two stainless steel contact) and it showed that there is little possibility for contact surface to have cold-welding [3]. However, when higher speed tracking, heat generation and wear problem will cause worse motion precision and bad control performance. In order to overcome these problems, special surface finish, DLC (Diamond Like Carbon) film coating, is to be done on those surfaces of the bearing component by sputtering deposition. The sputtered DLC film has high stiffness, high hardness, low friction coefficient and high atomic structural stability in vacuum.

PID Control Experiment

Figure 2 shows multistep control results. The control method is conventional PID control and the control gain is varied according to the motion

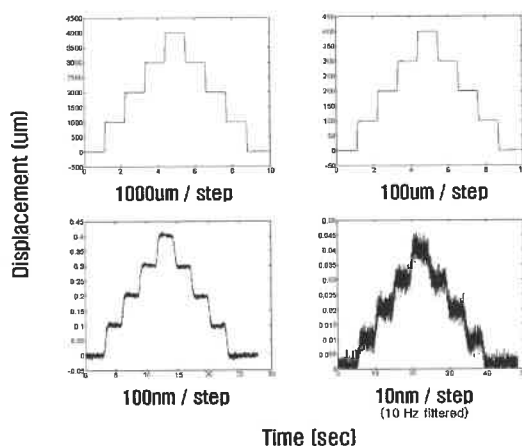
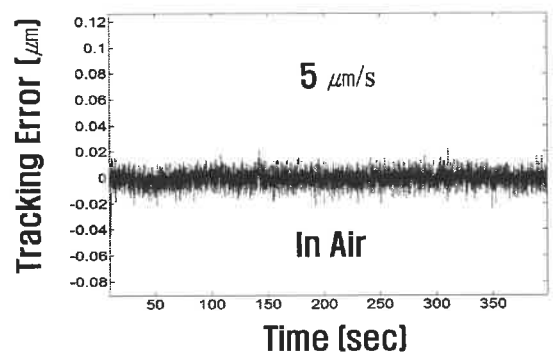
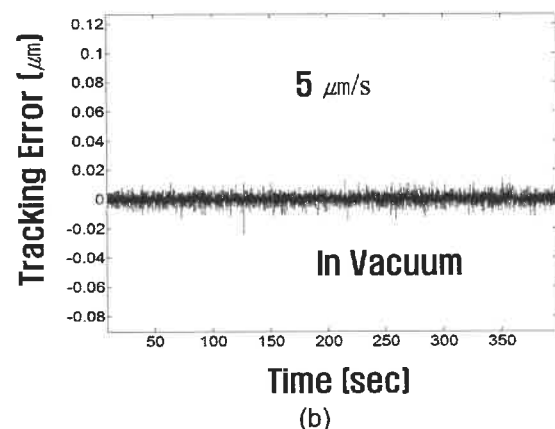


FIGURE 2. Multi-step control results at various single step amounts. A step control result as low as 10 nm is achieved.

step size as predicted. Many previous research results reported that physical models are different from each other when the movement range is above and below a certain value. When the moving range is long enough the control plant can be thought of as a system that has a mass and a damper. However, the system has an energy conservative model element, just like a spring component, when below a few micrometer movement and the controller gain need to be set higher. And in case of sub-micrometer movement the settling time increases because of abrupt model change which is a nonlinear burden to control system. The tracking control experiments are performed with various following speed in two environments of vacuum and ambient air. Figure 3 shows the result when 5 $\mu\text{m/s}$ speed tracking control was performed and somewhat better tracking performance can be observed in case of vacuum environment. With the experiment in vacuum the control gains needed to be



(a)



(b)

FIGURE 3. Tracking control experiment results : tracking speed is 5 $\mu\text{m/s}$. (a) in air (b) in vacuum

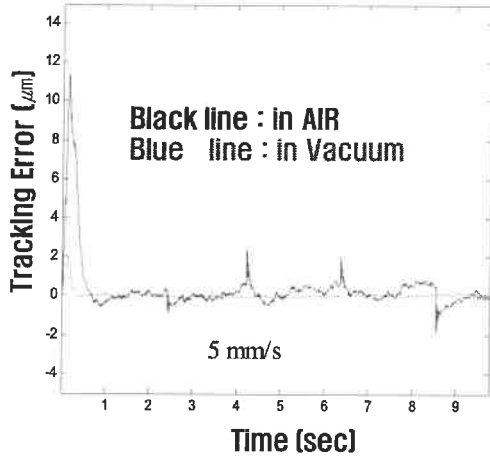


FIGURE 4. Tracking control experiment results: tracking speed is 5 mm/s in air and in vacuum.

increased higher possibly due to friction. A cold weld phenomenon could affect the control or dry contact condition could be a cause. When the tracking speed is about 5 mm/s the control performance is much worse in ambient air than that is in vacuum. This result can be explained by the existence of the adsorbed layer between guide rail and bearing surface in ambient air environment.

TRACKING CONTROLLER DESIGN : ZPETC+TDC

The TDC controller uses time delay estimation of the control plant model from the measured output. Therefore, the TDC is robust to plant parameter variation. The TDC controller makes the system follow the pre-determined reference model, which can be written as below [4],

$$\frac{d^2 e}{dt^2} + K_D \frac{de}{dt} + K_P e = 0 \quad (1)$$

With these K_D and K_P parameters one can define the reference model and generally critical damping condition is applied to the model. And the TDC digital controller can be written as follows,

$$u(k) = u(k-1) + \left[\left(\frac{\overline{M}K_D}{t_s} + \overline{M}K_P + \frac{\overline{M}}{t_s^2} \right) e(k) - \left(\frac{\overline{M}K_D}{t_s} + 2\frac{\overline{M}}{t_s^2} \right) e(k-1) + \frac{\overline{M}}{t_s^2} e(k-2) \right] \quad (2)$$

The $u(k)$ is control input at the k_{th} control event, t_s is the sampling time, $\overline{M}$ is the assumed coefficient value of the second order term of the system and $e(k)$ is the error input at the k_{th} control event. At the start of control, the initial conditions of those $(k-1)_{th}$ and $(k-2)_{th}$ values are set to zero. The Z-Transform result of the TDC controller is shown below,

$$C(z) = \left[\left(\frac{\overline{M}K_D}{t_s} + \overline{M}K_P + \frac{\overline{M}}{t_s^2} \right) z^2 - \left(\frac{\overline{M}K_D}{t_s} + 2\frac{\overline{M}}{t_s^2} \right) z + \frac{\overline{M}}{t_s^2} \right] \frac{1}{z^2 - z} \quad (3)$$

With the $C(z)$ controller and the plant model the closed loop system transfer function can be obtained. The ZPETC feedforward controller uses inverse model of the total closed loop system. The closed loop system transfer function can be written as equation (4).

$$G_{closed}(z^{-1}) = \frac{z^{-d} B_c(z^{-1})}{A_c(z^{-1})} \quad (4)$$

z^{-d} is a d-step plant delay. And the $G_{closed}(z^{-1})$ can have unstable zeros (out of unit circle in the Z complex domain) and ZPETC is designed with the following equation,

$$r(k) = \frac{A_c(z^{-1}) B_c^u(z)}{B_c^a(z^{-1}) [B_c^u(1)]^2} y_d(k+d) \quad (5)$$

$B_c(z^{-1})$ is divided into $B_c^a(z^{-1})$ and $B_c^u(z^{-1})$ and upper character u means unstable zeros and upper character a means stable zeros. In the denominator $B_c^u(1)$ means the DC gain value. The $y_d(k+d)$ is the desired position at $(k+d)_{th}$ control event and $r(k)$ is the input to the closed loop system. With the above ZPETC formula it has been theoretically proved that the ZPETC controller make the controlled system have zero phase following performance.

CONCLUSION

A vacuum compatible precision linear stage was designed and its control performance was investigated with conventional PID controller. And ZPETC+TDC controller was proposed for linear stage tracking in vacuum. The DLC coating on guide-rail is in process and with the designed ZPETC+TDC controller it is an ongoing procedure to investigate tracking control performance in high vacuum.

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PREPARATION OF TITANIUM NITRIDE – MOLYBDENUM DISULFIDE COMPOSITE THIN FILMS WITH LOW FRICTIONAL PROPERTIES

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INTRODUCTION

Titanium nitride (TiN) film is superior in hardness, adhesive strength to substrates and abrasion resistance. Therefore, the TiN film is used widely as a protective coating. However, friction coefficients of the TiN film are higher than those of diamond-like carbon (DLC) films [1]. In this study, the author tried to reduce the friction coefficient of sputtered TiN thin films by addition of molybdenum disulfide (MoS_2), which showed low frictional characteristics.

EXPERIMENTAL

Figure 1 shows the schematic diagram of DC magnetron sputtering apparatus with TiN and MoS_2 targets used for the TiN- MoS_2 composite films preparations. The TiN- MoS_2 composite films have been prepared on monocrystalline silicon, steel and aluminum substrates in Ar plasma. The input target power for TiN was fixed at 200 W. To control the amount of MoS_2 in the composite films and to investigate the influence of MoS_2 contents on the crystallinity, the hardness, the adhesive strength and the frictional properties of the films, the input target power for MoS_2 varied from 0 to 200 W at a constant substrate temperature of 350 °C. The residual pressure prior to deposition in the chamber was 2×10^{-3} Pa. The film depositions were carried out at an Ar gas flow rate of 50 sccm and under a work pressure around 2 Pa. The deposition time was varied to obtain the films with constant thickness. The film thicknesses were measured to be around 1 μm using a surface profilometer (DEKTAK IIA).

The chemical composition and bonding states of the TiN- MoS_2 composite films were investigated by X-ray photoelectron spectroscopy (XPS). In the XPS analyses, the binding energies of N 1s, Ti 2p, S 2p and Mo 3d were measured from the surface of as-deposited films without Ar^+ ion sputter cleaning, since Ar^+ ion bombardment decomposed the Mo-S bonding and reduced the sulfur concentration of the films. X-ray diffraction spectra were acquired at the X-ray incident

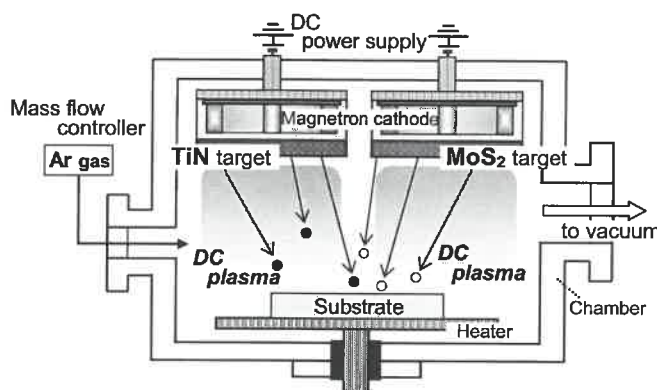


FIGURE 1. Schematic diagram of DC magnetron sputtering apparatus used for the TiN- MoS_2 composite films preparations.

angle 3° for evaluating the crystallinity of the TiN- MoS_2 composite films.

Nano-indentation hardness measurements were carried out with a trigonal (Berkovich) diamond indenter using a commercially available equipment (Nano Indentation Tester ENT-1100a: ELIONIX). Maximum indentation load was 5 mN. Scratch tests for evaluating the adhesion of the composite films to the substrates were made on a RESCA scratch tester CSR-02 device. The frictional properties of the TiN- MoS_2 composite films were examined using a ball on disk tester in an ambient atmosphere. An AISI52100 ball of 5 mm in diameter was slid against the films at a normal load of 0.5 N and a sliding speed of 30 mm/s.

COMPOSITION AND CRYSTALLINITY

The chemical composition and the Mo/Ti of the TiN- MoS_2 composite films are shown in Table 1. As increasing the target power for MoS_2 , molybdenum contents in the films, measured by X-ray photoelectron spectroscopy, increased up to 30 at.%. The XPS analyses revealed that Mo/Ti varied from 0.4 to 7.5 as the target power for MoS_2 increased. In the XPS N 1s spectra,

TABLE 1. Chemical composition of the TiN-MoS₂ composite films (at.%).

MoS ₂ target power	Concentration, at%					Mo/Ti
	Ti	N	Mo	S	O	
50W	19	22	8	7	44	0.4
100W	20	17	11	12	40	0.6
150W	11	12	20	24	33	1.8
200W	4	5	30	38	23	7.5

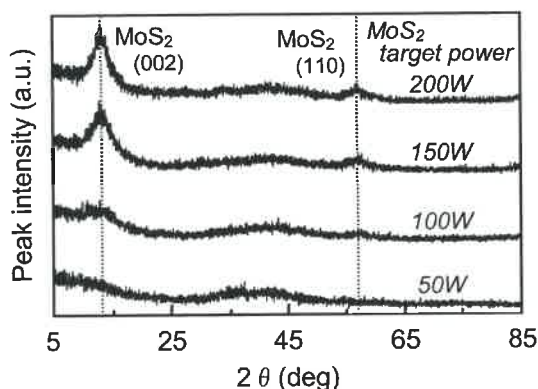


FIGURE 2. XRD spectra of the TiN-MoS₂ composite films prepared at various MoS₂ target powers.

a single peak existed at the binding energy of 390.0 eV. This peak is attributed to nitrogen bonded to titanium. As shown in Table 1, the titanium concentration was nearly equal to the nitrogen concentration. Therefore, titanium is thought to exist as TiN in the composite films.

Figure 2 illustrates XRD spectra of the composite films prepared at various MoS₂ target powers. It should be noted that no significant peaks assigned as crystalline TiN were found. Even in the XRD spectrum of the TiN-MoS₂ composite film prepared at the MoS₂ target power of 200 W, peaks corresponding to MoS₂ were weak and broad. Therefore, it is thought that the TiN-MoS₂ composite films obtained in this work had low crystallinity irrespective of MoS₂ content.

NANO-INDENTATION HARDNESS

Nano-indentation hardness of the TiN-MoS₂ composite films prepared at various MoS₂ target powers is shown in Figure 3. The TiN film had

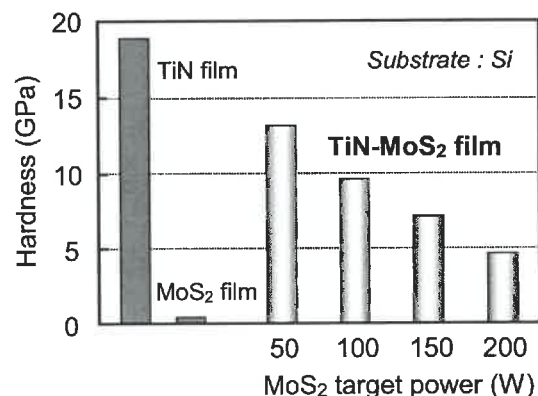


FIGURE 3. Nano-indentation hardness of TiN film, MoS₂ film and the TiN-MoS₂ composite films prepared at various MoS₂ target powers.

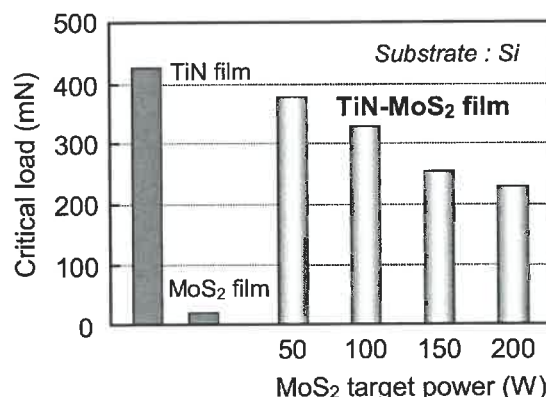


FIGURE 4. Critical loads of TiN film, MoS₂ film and the TiN-MoS₂ composite films prepared at various MoS₂ target powers.

high hardness of 19 GPa, and the hardness of MoS₂ film was 0.1 GPa. As the MoS₂ target power increased, the nano-indentation hardness of the composite films decreased. However, the hardness of the composite films prepared at the MoS₂ target power of 200 W, in which Mo/Ti was as high as 7.5, was about 5 GPa and was much higher than that of the MoS₂ film.

ADHESIVE STRENGTH

Figure 4 shows critical loads in scratch tests for the TiN film, the MoS₂ film and the TiN-MoS₂ composite films prepared at various target power for MoS₂. The critical load of the TiN-MoS₂ composite film prepared at the MoS₂ target power of 50 W was about 400 mN, which was almost equal to that of the TiN film. The adhesive strength of the TiN-MoS₂ composite

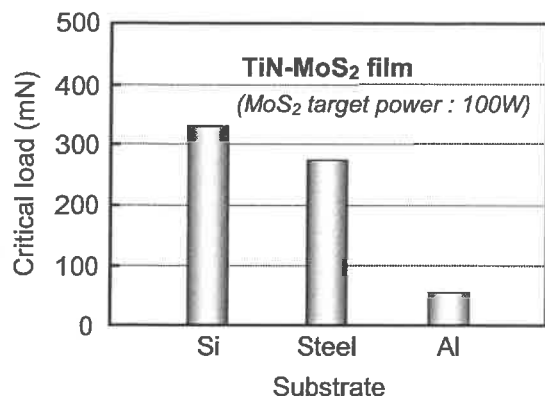


FIGURE 5. Critical loads in scratch tests for the TiN-MoS₂ composite films prepared at the MoS₂ target power of 100 W on various substrates.

films decreased from 400 mN to 220 mN with increasing the MoS₂ target power. However, the adhesive strength of the TiN-MoS₂ composite films was much higher than those of the MoS₂ films even when the MoS₂ concentration in the composite film was above 80 at%.

The critical loads in the scratch tests for the TiN-MoS₂ composite films prepared at the MoS₂ target power of 100 W on various substrates are summarized in Figure 5. The critical load of the composite film prepared on the steel substrate was close to that on the silicon substrate. On the other hand, the composite film on the aluminum substrate had very weak adhesive strength.

FRICITIONAL PROPERTY

Figure 6 indicates the friction coefficients of the TiN-MoS₂ composite films prepared on the silicon substrates and a DLC film against a AISI52100 ball at room temperature. While the friction coefficient of the composite film prepared at the MoS₂ target power of 50 W was around 0.7, that of the film prepared at the MoS₂ target power of 100 W was stable at 0.1 in dry condition, which was almost equal to those of the DLC films and the MoS₂ films [2,3].

The friction coefficients of the TiN-MoS₂ composite films prepared on the silicon substrates and the DLC film measured at 200 °C are shown in Figure 7. Even at 200 °C, the TiN-MoS₂ composite film prepared at the MoS₂ target power of 100 W exhibited the good tribological performance with a low friction coefficient below 0.1, which was smaller than that of the DLC film as shown in Figure 7.

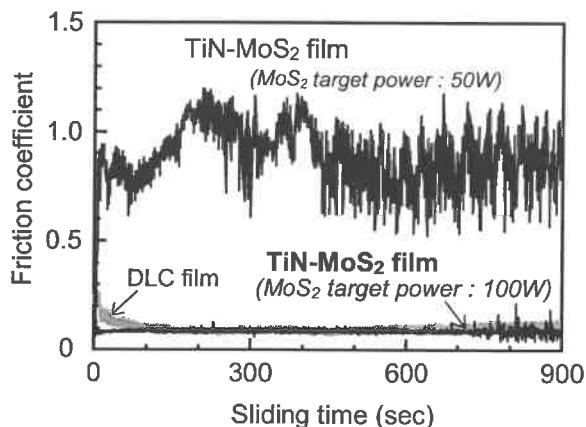


FIGURE 6. Friction coefficients of the TiN-MoS₂ composite films on silicon substrate and DLC film at room temperature in dry condition.

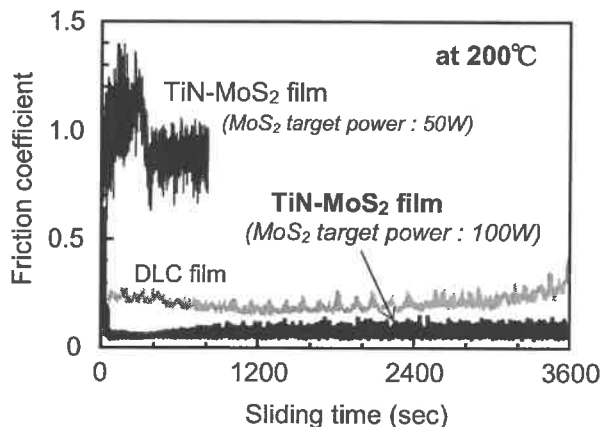


FIGURE 7. Friction coefficients of the TiN-MoS₂ composite films on silicon substrate and DLC film at 200 °C in dry condition.

Figure 8 indicates the friction coefficients of the TiN-MoS₂ composite films prepared at the MoS₂ target power of 100 W on steel substrate and on aluminum substrate. The composite film on the steel substrate had the excellent frictional properties as well as the films on the silicon substrate. On the other hand, the friction coefficient of the composite film prepared on the aluminum substrate was as high as 0.5.

Figure 9 shows the wear tracks on the TiN-MoS₂ composite films prepared at the MoS₂ target power of 100 W on the steel substrate and on the aluminum substrate. The composite film on the steel substrate was slightly worn off. On the other hand, apparent wear was observed on

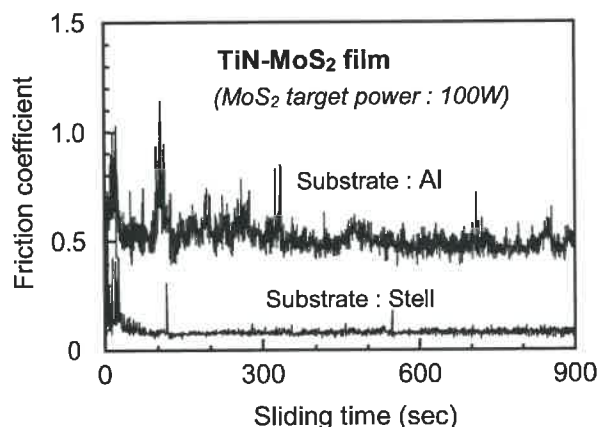


FIGURE 8. Friction coefficients of the TiN-MoS₂ composite films on steel substrate and on aluminum substrate at room temperature in dry condition.

the wear track of the composite film on the aluminum substrate. This is partly due to the low adhesive strength of the composite film on the aluminum substrate as shown in Figure 5.

FUTURE WORK

In this study, an obvious layered structure of MoS₂ was not observed in the composite films. Therefore, high concentration of MoS₂ (high Mo/Ti) was necessary for improving the frictional properties of the TiN films. However, the film formation process mentioned above is thought to be one of the film design method for obtaining the hard and lubricious films. By developing the layered structure of MoS₂ and improving the crystallinity of TiN, the TiN-MoS₂ composite films will be rather effective for reducing the friction and wear of the sliding parts, such as the apex seal of the rotary engine.

ACKNOWLEDGEMENTS

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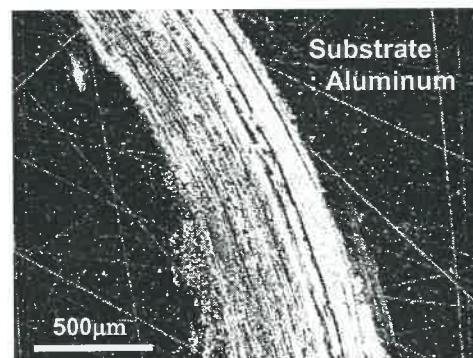
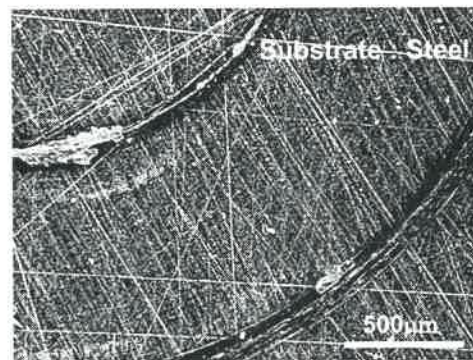


FIGURE 9. Wear tracks on the TiN-MoS₂ composite films prepared on steel substrate and on aluminum substrate after sliding tests of 900 sec at room temperature.

FEMTOSECOND LASER ABLATION OF SAPPHIRE

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INSTRUCTION

Sapphire is widely used in the field of optical components and micromechanical devices owing to its useful mechanical, optical, and electrical properties. However, sapphire is mechanically and chemically difficult to machine because of its high hardness and chemical stability. The difficulty of machining sapphire has restricted the development of advanced device structures. As sapphire is inert to most types of wet chemical and dry etching, laser ablation has been proposed as a potential machining method. The interaction of laser pulses with sapphire materials has been investigated for many years [1-3]. However, a thermal effect is unavoidable, even in ultraviolet laser processing. Recently, the femtosecond laser ablation of sapphire has been attracting much attention because femtosecond laser ablation can produce precise, well-defined micrometer-sized structures in materials that cannot be processed with nanosecond pulsed lasers. A number of studies on the femtosecond laser ablation of sapphire have been carried out [4-7], and a mechanism for the femtosecond laser ablation of sapphire has been proposed [8-10].

The advanced development of III-V nitride devices such as those of GaN and InGaN [11,12] has prompted research into a suitable method of patterning sapphire. However, further experiments need to be carried out to obtain fine structures on sapphire. In this study, experiments on the femtosecond laser ablation of sapphire were carried out focusing on several aspects including the investigation of laser-induced periodic surface structures (LIPSS) on sapphire, and the relationship between ablation depth and the number of pulses, and a method of improving the surface quality of sapphire. As a result of our investigations, a sample of sapphire with parallel microgrooves on the surface was fabricated.

EXPERIMENTS

In the experiments, we used a commercially available amplified Ti:sapphire laser system that generated 164 fs laser pulses with a maximum pulse energy (E_p) of 1 mJ at a 1 kHz repetition rate and with a central wavelength of $\lambda = 780$ nm. The laser beam had a Gaussian profile with a diameter of 6 mm. The incident laser beam was irradiated onto a sapphire sample through a 4 mm aperture positioned normal to the sample and a 10× or 50× microscope objective lens. The sample was placed on a scanning stage and translated at various scanning speeds (v) under computer control during irradiation. The scanning direction was perpendicular or parallel to the laser polarization direction. All experiments were carried out in air at atmospheric pressure and room temperature. After irradiation, the sample was rinsed for 30 minutes with acetone in an ultrasonic cleaner to remove any debris from the ablation process. The morphology of the structures was examined by scanning electron microscopy (SEM). The sample was coated with a gold layer of approximately 22 nm thickness to ensure conduction in the SEM observation. The surface profile was measured by atomic force microscopy (AFM).

RESULTS AND DISCUSSION

The sapphire was first irradiated by different pulse energies and numbers of laser pulses under static irradiation, and then the fabrication of microgrooves on sapphire by femtosecond laser pulses was investigated through line-scanning experiments.

Laser-Induced Periodic Surface Structures on Sapphire

Figure 1 shows an image of LIPSS formed on the sapphire surface when it was irradiated with a pulse energy close to the ablation threshold

and 20-50 laser shots. The spatial period of LIPSS on the sapphire surface was approximately 340 nm. The orientation of the LIPSS was perpendicular to the polarization direction of the laser beam.

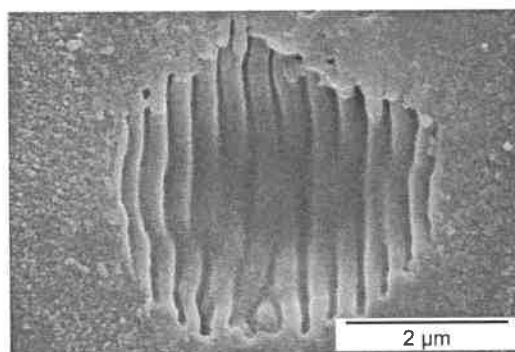


FIGURE 1. SEM image of LIPSS formed on sapphire by femtosecond laser pulses with $E_p = 1.95 \mu\text{J}$ and 50 laser shots.

The formation of LIPSS on sapphire was caused by interference between the laser-induced plasma and the incident laser beam [13]. Further investigations on the fine nanostructures formed on sapphire by femtosecond laser pulses similar

to those that Shinoda et al. obtained on diamond [13] are being carried out because of their potential application to nanotechnology.

Laser Processing of Sapphire with Different Pulse Energies and Laser Shots

Upon irradiation with ultraviolet and ultrashort laser pulses, two ablation phases can be observed with increasing number of laser pulses [2,4,5]. One is a 'gentle' ablation phase characterized by a low rate of material removal, which has been attributed to particle vaporization. The second phase is characterized by a high rate of material removal and the production of molten droplets, which has been attributed to a phase explosion [2,4,5]. In our experiment, the same results were obtained, in agreement with previous works by Tam et al. [2] and Ashkenasi et al. [5]. Figure 2 shows AFM images of the cross-sectional profile of the crater produced by the irradiation of femtosecond laser pulses with a pulse energy of $3.64 \mu\text{J}$ and different number of laser pulses. During the first 10 pulses, the laser-ablated area is generally smooth, as shown in Figures 2(a), 2(b), and 2(c). After 10 laser shots, rough regions start to form in the ablated area, as shown in Figure 2(d).

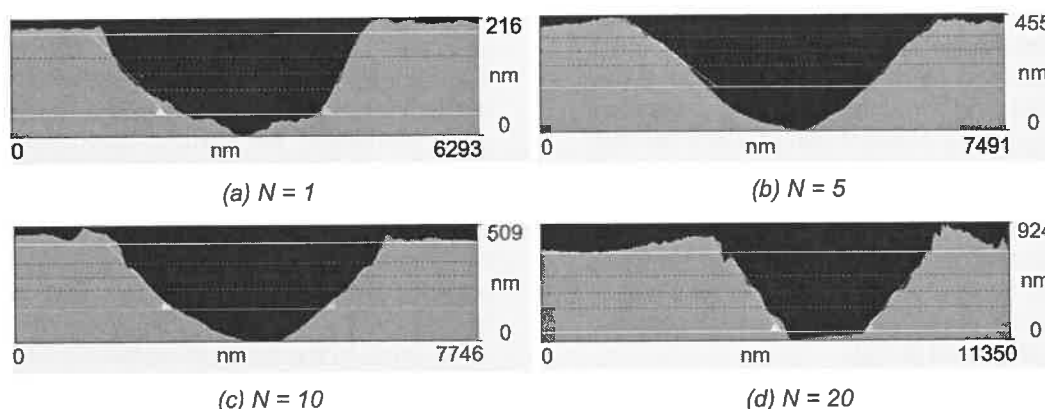


FIGURE 2. AFM images of cross-sectional profile of the crater produced by irradiation of femtosecond laser pulses with $E_p = 3.64 \mu\text{J}$ and different numbers of laser pulses.

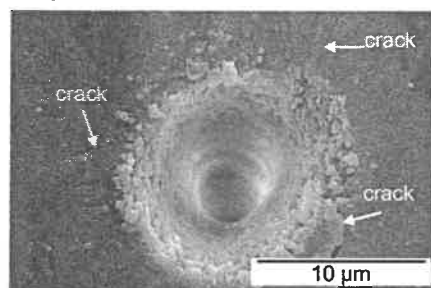
Cracks have a negative effect on the applicability of sapphire. The formation of cracks is unavoidable, even in ultraviolet laser processing with long pulses, because of the thermal effect [1]. The sapphire ablated by femtosecond laser pulses is free of cracks under appropriate irradiating conditions. Figure 3 shows SEM images of the craters produced by the irradiation of femtosecond laser pulses with 1000 laser shots and different pulse energies. When the pulse energy was high, cracks were observed around the ablated crater, as shown in

Figure 3(a). When the pulse energy decreased, no cracks were observed around the ablated crater, as shown in Figure 3(b).

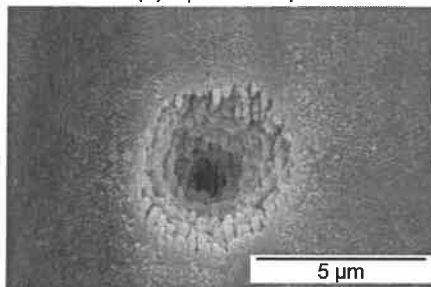
Variation of Ablation Depth with Pulse Energy and Number of Laser Pulses

The ablation depth of the crater is an important parameter for investigating the mechanism of laser ablation and for expanding the range of applications of sapphire. In our experiment, sapphire was first irradiated by femtosecond laser pulses that were propagated through a $50\times$

microscope objective lens. The dimensions of the crater were measured by AFM and the ablation depth of the crater was obtained by measuring its cross-sectional profile. The ablation depth as a function of the number of laser pulses is illustrated in Figure 4 for three different pulse energies. The ablation depth of the crater increased with the number of laser pulses up to 100 shots. Above 100 laser shots, the ablation depth remained almost unchanged. The maximum obtainable depth was fixed. The mechanism of the femtosecond laser ablation of sapphire involves the phase explosion [2,5]. Ashkenasi et al. analyzed in detail the occurrence of the phase explosion during the femtosecond laser ablation of sapphire [5]. However, their results were puzzling in that the depth of ablation should be independent of the laser fluence if phase explosion is the main process leading to the removal of materials [5]. Our results may partially clarify this anomaly. Figure 4 shows that the ablation depth of the crater was independent of the pulse energy when the number of laser pulses was smaller than 25. As the number of laser pulses increased to 100, the ablation depth remained almost the same. The results showed that the ablation depth was independent of the laser pulse energy provided it was sufficiently high for phase explosion to occur.



(a) $E_p = 16.49 \mu\text{J}$



(b) $E_p = 1.33 \mu\text{J}$

FIGURE 3. SEM images of the craters produced by irradiation of femtosecond laser pulses with 1000 laser shots and different pulse energies.

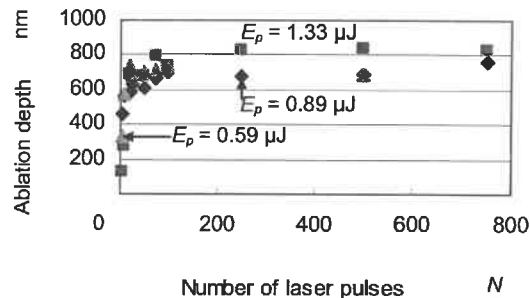
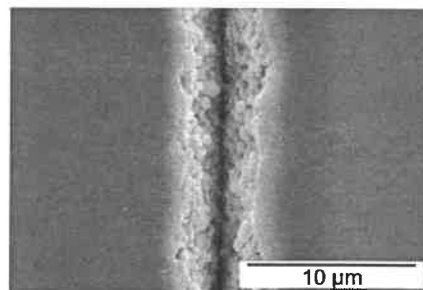


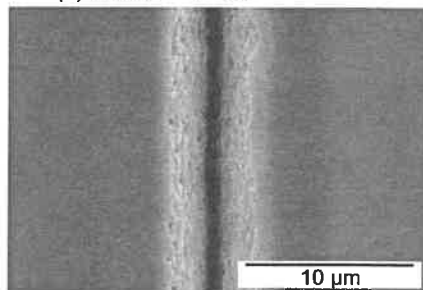
FIGURE 4. Relationship between the ablation depth of the crater and number of laser pulses as a parameter of pulse energy ($50\times$ objective lens).

Microgroove Fabrication on Sapphire by Femtosecond Laser Pulses

First, experiments to determine the dependence of the surface quality of the ablated crater on the crystal plane of the sapphire were carried out. Three types of crystal plane, c-plane, a-plane, and r-plane, was considered. The experiments were carried out with different pulse energies and number of laser pulses. The results showed that the surface quality of the ablated crater was higher when femtosecond laser pulses were irradiated on the c-plane, resulting in the fabrication of microgrooves on c-plane sapphire.



(a) Number of laser scans = 2



(b) Number of laser scans = 10

FIGURE 5. SEM images of microgrooves fabricated with different numbers of laser scans by femtosecond laser pulses with $E_p = 1.64 \mu\text{J}$ and $v = 100 \mu\text{m/s}$.

Second, experiments on the fabrication of microgrooves on sapphire by femtosecond laser pulses were carried out under various conditions of pulse energy, scanning speed, and number of laser scans. The surface quality of the microgrooves was improved when the number of laser scans increased, as shown in Figure 5, although as the number of laser scans increased, the ablation depth of the microgrooves remained almost the same. The finding that the ablation depth of the microgrooves was independent of the number of laser scans was consistent with the results of the static irradiation experiments shown in Figure 4.

Finally, a sample of parallel microgrooves with high surface quality was fabricated on sapphire by femtosecond laser pulses, as shown in Figure 6.

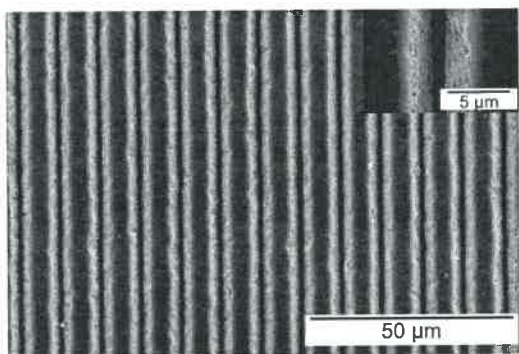


FIGURE 6. SEM image of parallel microgrooves on sapphire fabricated by femtosecond laser pulses with $E_p = 1.64 \mu\text{J}$, $v = 100 \mu\text{m/s}$, and 10 laser scans.

CONCLUSION

Experiments on the femtosecond laser ablation of sapphire were carried out. The main conclusions are as follows.

- 1) Nano- and microstructures were fabricated on sapphire.
- 2) The craters formed by laser ablation are free of cracks under appropriate irradiating conditions. The removal of materials by femtosecond laser pulses was mainly caused by phase explosion.
- 3) The finding that the ablation depth of the crater was independent of the laser pulse energy clarified the puzzling phenomenon reported by Ashkenasi et al. that the depth of ablation should be independent of the laser fluence if phase explosion is the main process leading to the removal of materials.

- 4) The surface quality of the ablated crater was higher when femtosecond laser pulses were irradiated on c-plane sapphire.
- 5) In the line-scanning experiments, the surface quality of the ablated microgrooves fabricated by femtosecond laser pulses was improved by increasing the number of laser scans.

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HIGH-ACCURACY ATOMIC FORCE MICROSCOPE FOR DIMENSIONAL METROLOGY (II)

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INTRODUCTION

We are designing and testing a high accuracy AFM (HAFM) to be integrated with the Sub Atomic Measurement Machine (SAMM) and be used for dimensional metrology [1]. The SAMM, which has been developed at the University of North Carolina at Charlotte, will provide the sample raster within a 25-mm by 25-mm range with 1-nm repeatability and 25-nm accuracy [2]. The HAFM is designed to track the sample surface with better than 1nm repeatability. This paper focuses on the HAFM.

The HAFM uses a self-sensing and self-actuating Akiyama[®] AFM probe [3], which eliminates the need for an optical lever sensing mechanism and results in a more compact AFM head for the SAMM stage. The AFM probe

operates in controlled-amplitude self-resonance mode to achieve relatively high bandwidth. Utilizing monolithic flexures and a piezoelectric actuator, HAFM has a 20- μ m tracking-motion range. The HAFM's metrology frame is separated from its force frame and is constructed from INVAR to reduce error due to mechanical deformations and temperature variations. To measure the probe motion, the HAFM uses three ADE8810 capacitive displacement sensors to directly measure the tracking motion with 0.22-nm RMS noise in a 1-kHz 3dB bandwidth.

HAFM DESIGN

The following sub-sections present the HAFM design.

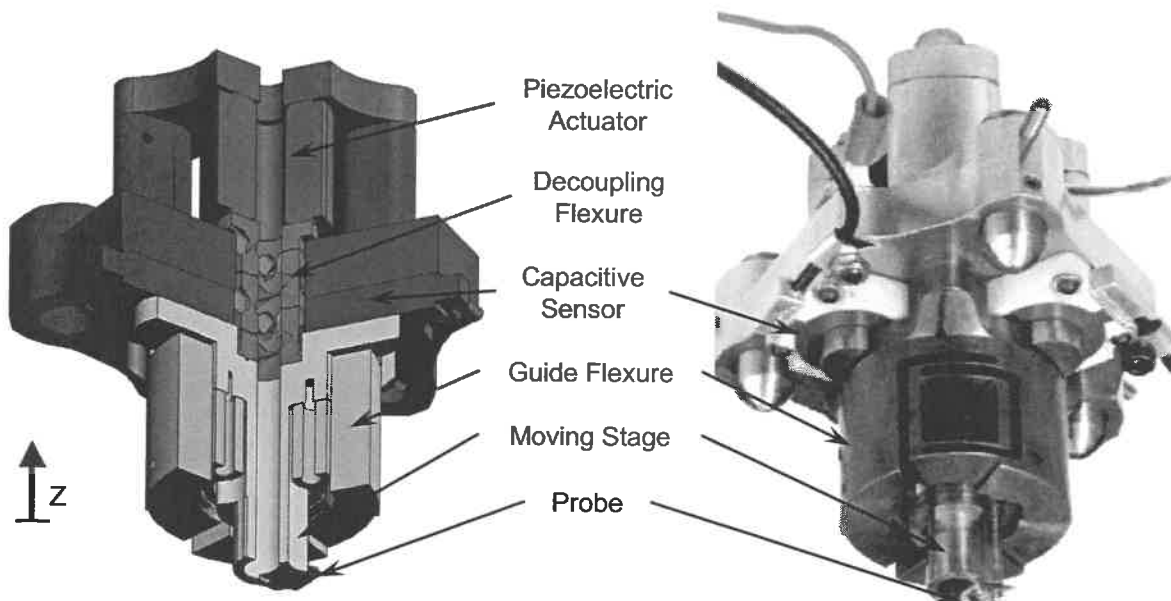


FIGURE 1. CAD cross-sectional view (left) and photo (right) of assembled HAFM head.

Self-Sensing AFM Probe

The most common AFM probe configuration uses an optical-lever sensing mechanism where a laser beam is reflected off the probe. The thermal noise introduced by the laser beam heating close to the measurement point can cause significant measurement error [3,4]. We use a self-sensing Akiyama® probe to eliminate the laser. The probe consists of a U-shaped AFM cantilever, which is symmetrically attached to the end of a quartz tuning fork [3]. The tuning fork is used for both driving the cantilever and for sensing the tip-sample interactions. Applying a harmonic excitation voltage to the tuning fork close to its resonance creates a harmonic vertical motion of the cantilever tip. Simultaneously, the electrical admittance of the tuning fork, which varies depending on the tip-sample interaction, can be used as feedback.

Given its tuning fork design, the probe has a high quality-factor (500-1500 in air) meaning that it dissipates little energy per cycle. As a result, transients in the probe's open-loop response disappear relatively slowly and will correspondingly reduce the bandwidth of sensing if the probe's open loop amplitude response is used as feedback. References [4-8] have overcome this limitation by using the phase response in closed-loop. The excitation frequency is controlled to maintain constant phase shift between the probe's voltage and current. The varying excitation frequency can then be used as feedback on tip-sample distance. References [4,7] control the excitation frequency in closed-loop using a direct measurement of the probe's phase response. Alternatively, [5,6] use a phase-shifted positive-feedback scheme where the probe automatically self-resonates at a frequency fixed relative to its moving open-loop phase response. Our HAFM uses a design similar to [5,6] where the probe is set in self-resonance with controlled amplitude at a frequency set by constant probe phase shift.

Monolithic Flexural AFM Head

The HAFM head moves the AFM probe perpendicular to the sample (z-axis) to maintain a constant excitation frequency, and thus a fixed tip-sample distance. The head is designed for 20- μm z-axis range with sub-nanometer resolution. The completed HAFM head and a CAD cross-sectional view of the head are shown in figure 1.

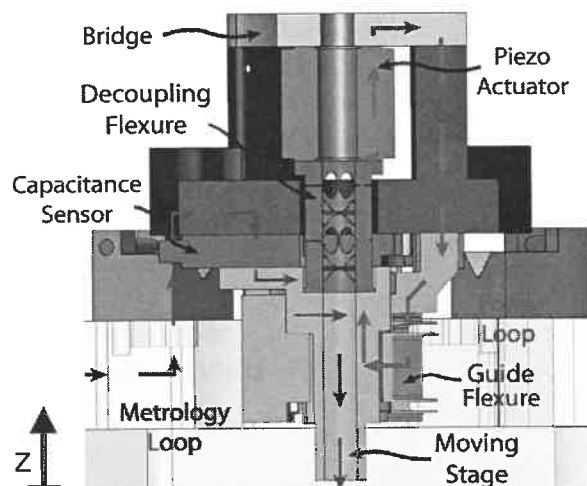


FIGURE 2. Cross-sectional view of HAFM with overlaid force and metrology loops.

Two monolithic flexures are utilized in the head design. The guide flexure is used to constrain the motion to the z-direction. It uses a double-back or crab-leg design so that the parasitic radial motions of the flexure cancel out. The decoupling flexure connects the piezoelectric actuator to the moving stage. It is designed to be stiff in the z-direction and flexible in the remaining degrees-of-freedom so that the piezoelectric actuator's error motion is attenuated and not significantly transmitted to the moving stage.

To avoid measurement errors caused by mechanical deformation, the metrology and the force loops are separated. Figure 2 shows the HAFM's separated force and metrology loops. To minimize errors due to thermal expansion, the metrology-loop's components are manufactured from INVAR. Once integrated with SAMM, HAFM's operating environment will be temperature controlled to within 0.01 degrees Celsius by the SAMM enclosure and environmentally controlled room.

Control and Instrumentation

An overall picture of HAFM's control and instrumentation design is shown in figure 3. The self-sensing Akiyama probe along with its controlling electronics are used to sense the tip-sample distance. The head-controller tracks the surface by maintaining a constant tip-sample distance. The tracking motion is concurrently captured by three capacitive probes and is recorded as the sample height profile.

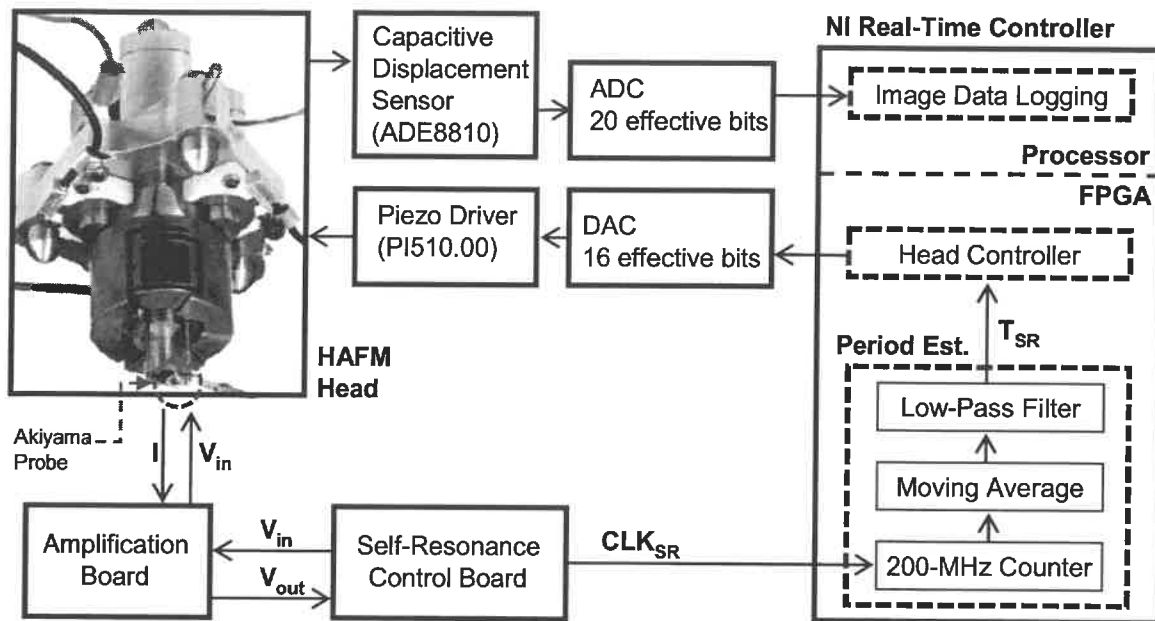


FIGURE 3. Overall configuration of HAFM control and instrumentation.

The self-sensing probe's output is a harmonic current signal with up to 500 nA amplitude. For better noise performance, an amplification board is located close to the probe and is used to convert and amplify the probe current signal (I) into a voltage signal (V_{out}). The probe controller board sets the probe in self-resonance with controlled amplitude and uses a comparator to convert the harmonic probe signal into a self-resonance clock (CLK_{SR}). This self-resonance controller configuration is similar to that in [9]. The period estimation module, which is implemented within the real-time controller's FPGA module, measures the period (T_{SR}) of the CLK_{SR} . The period measurement is used as feedback to control the tip-sample distance. We use high-resolution ADCs, with 20.1-effective-bit resolution, and DACs, with 16-bit resolution, which were designed by Gawlik and Otten [10]. A PI510.00 amplifier drives the piezoelectric actuator. Three ADE8810 capacitive sensors are averaged digitally to measure the head tracking motion with 0.22-nm RMS noise in a 1-kHz 3dB bandwidth.

Self-resonance period measurement and head control are performed using the FPGA. A 200-MHz counter measures the time between the falling and rising edges of CLK_{SR} . For increased resolution a moving average and a low-pass filter are applied to the period measurements.

The head controller compensates and closes the surface tracking loop.

EXPERIMENTAL RESULTS

The experimental results, presented below, are obtained with the HAFM operating in the self-resonance mode. The tests have been conducted in a room environment without any temperature control and are only to evaluate HAFM's z-axis tracking performance before integration with the SAMM stage.

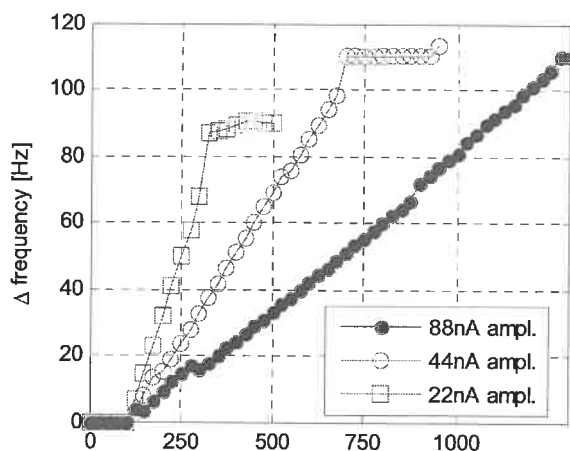


FIGURE 4. Change in self-resonance frequency vs. tip-sample gap for different oscillation amplitudes.

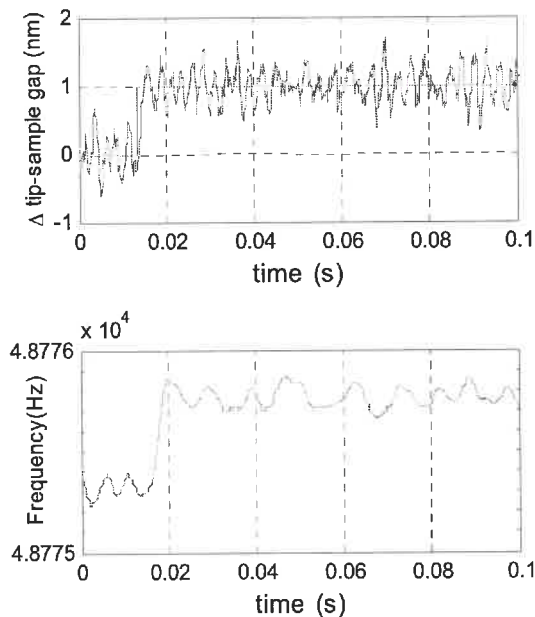


FIGURE 5. HAFM's step response: 1-nm step generated using cap probe feedback (Top); AFM probe's corresponding response (Bottom).

The self-resonating AFM probe's response to the tip-sample variation is shown in Figure 4. The tip-sample distance was controlled based on the capacitive probes' feedback. The AFM probe's sensitivity is found to be inversely proportional to its oscillation amplitude. However, at lower amplitudes, the probe seems to stick to the surface and also its signal to noise ratio is reduced. The oscillation amplitude of 20-nA is selected as a good trade-off for tests within the room environment.

Figure 5 shows a 1-nm step created by the head using the capacitive probes' feedback and the corresponding AFM probe's response. The capacitive probes' measurement shows 0.22nm RMS noise. Operating with 20-nA amplitude, the 1-nm step is detected by the probe with RMS noise of 0.052 Hz, which is equivalent to 0.12 nm. For this measurement, a 32-point wide moving average at the 98 kHz edge rate and a 4th order Butterworth filter with 150-Hz -3dB frequency are used. Electrical noise, thermal variation, and mechanical vibration are predicated as the main factors limiting the probe's resolution.

CONCLUSION

We have designed and implemented a high accuracy AFM (HAFM) for 20- μ m range of motion with sub-nanometer resolution. Head

position control and sensing, with 0.22nm RMS noise, using the capacitive sensors' feedback, and tip-sample gap detection, with 0.12nm RMS noise, using the self-sensing probe have been demonstrated experimentally.

Currently, we are developing electronics to improve the AFM probe's signal to noise ratio. In the near future, The HAFM's position control loop will be closed using the AFM probe's feedback. In this configuration, the low pass-filter of figure 3 will be effectively replaced by the loop compensator and the mechanical plant's dynamics. The HAFM will be integrated with the Sub Atomic Measurement Machine (SAMM) to be tested for sample profile measurement. The HAFM's performance is expected to be improved once tested within SAMM's hermetic enclosure with temperature control and vibration isolation.

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DESIGN OF MICRO-SCALE MULTI-AXIS FORCE SENSORS FOR PRECISION APPLICATIONS

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SYNOPSIS

In this paper, we present a design process created to support the engineering of precision, multi-axis MEMS force sensors. A closed form model is presented which optimizes the strain sensitivity of MEMS force sensors. This model uses flexure geometry and system architecture to determine how noise propagates through the sensor system and to estimate the sensor's accuracy and resolution. Using this model, a three degree-of-freedom (DOF) force sensor was designed with the intent to measure the adhesion forces between biologically active surfaces with nN-level resolution and accuracy.

INTRODUCTION

Multi-axis force sensing is required in biology, materials science and nanomanufacturing. Unfortunately, few fine-resolution, multi-axis MEMS force sensors exist because of the design and manufacturing challenges encountered when creating them. Little work has been conducted to enable researchers/practitioners to be able to model the limits of sensor performance. The lack of this knowledge makes it difficult to determine when improvements in performance outweigh the associated design and manufacturing challenges. Herein we present a sensor design process that incorporates closed-form modeling of the sensing system. The process makes it feasible to design small-scale, precision, multi-axis force sensors.

MEMS force sensors tend to rely on one of three sensing methods: capacitive sensing, optical laser detection, and piezoresistive sensing. Several multi-axis force sensors have been developed using capacitive sensors [1,2]. These sensors are difficult to fabricate and require relatively large sensor areas (mm^2) for each axis in order to achieve high force resolution. This makes capacitive sensing impractical for small, inexpensive, multi-axis force sensors. Optical sensors are widely used in atomic force microscopy (AFM) to make high resolution force

measurements in one axis. Optical sensors are rarely used in applications of interest due to the difficulty and cost of integrating multiple sets of optics into a small region. Also, optical sensors require relatively large lasers which make it impossible to miniaturize the force sensing system to the micro-scale. Piezoresistive sensors offer the most promise at the micro-scale due to their small size and relative ease of integration into MEMS devices. Piezoresistive transducers are commonly found in MEMS devices such as pressure sensors, accelerometers, and AFM cantilevers [3]. Several dual-axis MEMS cantilevers with nN-level resolution have previously been demonstrated [4,5]. The modeling and design work in this paper will focus on piezoresistive sensing techniques.

CLOSED FORM MODELING OF MULTI-AXIS MEMS FORCE SENSOR

In order to optimize the design of MEMS force sensors, a closed-form model was created with the aim of tracking the propagation of the force signal and various noise sources, offsets, and thermal drifts through the sensor system. The model provides insight into (i) which parameters have the largest effect on the signal-to-noise ratio and (ii) the sensitivity of the sensor's resolution to relevant noise sources. In addition, this model is used to maximize the accuracy of the system by evaluating the system's sensitivity to offset biases and thermal errors.

Flexure Design

As may be seen in Figure 1, the force input is translated into strain through the flexure system. For a fixed-guided beam, this translation is given by the parameter F_F in Eq. 1, where ε is the strain, F is the force at the guided end of the beam, L is the beam length, b is the beam width, h is the beam thickness, and E is the elastic modulus.

$$F_F = \frac{\varepsilon}{F} = \frac{3L}{Ebh^2} \quad (1)$$

The beam geometry is set so that the flexure

reaches $1/3$ of its yield stress at maximum force.

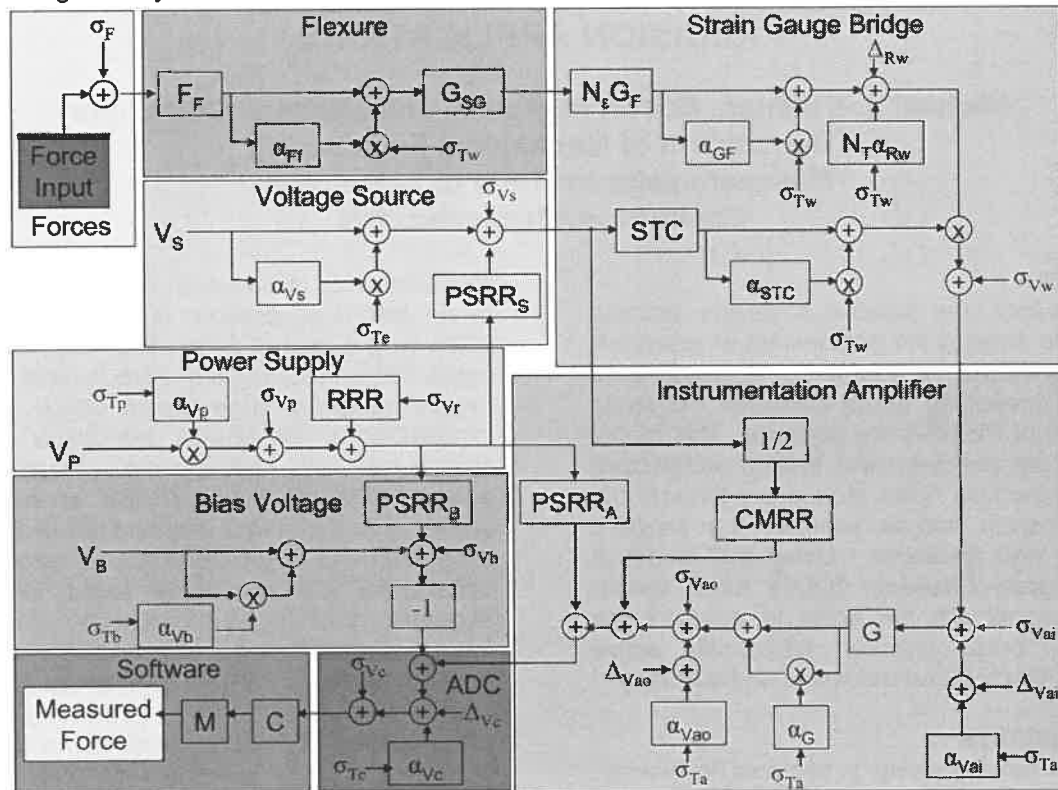


FIGURE 1. Model of signal and noise flow through the force sensor system

Noise Modeling and Offsets

Once the force is translated into a strain, and the effects of thermomechanical noise (σ_F), thermal drifts (σ_{Tw}), and resistor placement (G_{SG}) on the strain reading have been accounted for, the strain reading is fed into a Wheatstone bridge. The Wheatstone bridge converts the strain into a voltage using the gauge factor (G_F) and bridge number (N_e). Noise from the power supply and bridge voltage source propagates into the Wheatstone bridge through the span temperature compensation (STC). The STC is used to cancel the effects of temperature fluctuations on the gauge factor of the piezoresistors (α_{GF}). Thermal fluctuations can also create a bias in the bridge by causing the resistivity of the resistors in the bridge to change unevenly. This effect can be minimized ($N_T \rightarrow 0$) by placing the resistors close together so that they see essentially the same temperature change. There is also often an offset in the bridge (Δ_{Rw}), due to fabrication variations between the resistors in the bridge, which can cause accuracy errors.

While some noise is passed into the Wheatstone bridge via the power supply, voltage source, and

flexure system, most of the noise in the Wheatstone bridge is a result of the piezoresistors. There are two main sources of noise in the piezoresistors: (1) Johnson noise created by the thermal agitation of electrons in a conductor and (2) flicker noise caused by conductance fluctuations that manifest during the capture and release of charge carriers in the piezoresistor [6]. The total Wheatstone bridge noise that results from the two noise sources is given in Equation (2), where k_b is the Boltzmann constant, T is the temperature, R is the bridge resistance, f is the frequency in Hz, α is the Hooke constant, N is the number of charge carriers, and V_s is the voltage across the resistors.

$$\sigma_{Vw} = \sqrt{4k_bTR(f_{\max} - f_{\min}) + \frac{\alpha V_s^2}{N} \ln\left(\frac{f_{\max}}{f_{\min}}\right)} \quad (2)$$

The signal and the noise from the Wheatstone bridge are fed into the instrumentation amplifier where they are multiplied by a gain (G) and additional noise (σ_{Vai} , σ_{Vao}), offsets (Δ_{Vai} , Δ_{Vao}), temperature sensitivities (α_{Vai} , α_{Vao} , α_G) and a bias are added into the signal. In addition, noise from the power supply and voltage source are fed into the system through the amplifier. The

signal is then fed into the analog-to-digital converter (ADC) where it is digitized and passed into the software. Noise (σ_{vc} , σ_{vb}), offsets (Δ_{vc}) and temperature sensitivities (α_{vc} , α_{vb}) are added to the signal at the ADC from both the ADC itself and the voltage bias. In the software, the digitized signal is converted into a force measurement through conversion via calibration, C , and coordinate transform, M , coefficients.

DESIGN OF A THREE AXIS FORCE SENSOR

This type of multi-axis, precision force sensing is needed to measure adhesion forces between cells and various material surfaces. For example, these measurements enable one to know (i) how well cells bond to different types of biomedical implant materials and (ii) the effects that drug coatings have on the prevention/promotion of adhesion. This type of measurement is necessary where the mechanical properties of the cell-implant interface are critical [7].

Functional Requirements

Accurate measurement of cell adhesion forces between surfaces requires multi-axis sensing to make sure that the surfaces are suitably positioned and oriented, thereby ensuring that load is applied evenly over the surfaces during testing. Cellular adhesion forces are typically on the scale of nN [8]. When cells are arrayed on the surface being tested, the adhesion force is on the order of 100's of μ N. To be useful, our force sensor must have 100's of μ N range, nN resolution, and be capable of measuring forces in a direction parallel to the normal to the plane of contact (Z) and torques about axes that are parallel to the planes of contact, (θ_x , θ_y).

Sensor Design

The noise flow model presented in this paper was used to design a 3-axis forces sensor that is intended to satisfy functional requirements for measuring cell adhesion forces. This force sensor was designed to fit on top of a HexFlex nanopositioner system [9]. This will make it possible for the force sensor to be accurately positioned/oriented while minimizing the cost of the total system. Figure 2 shows the sensor, which is comprised of three coplanar flexures with integrated piezoresistors at the base of the flexures. The piezoresistors are in a $\frac{1}{2}$ bridge arrangement. A full Wheatstone bridge was not used due to fabrication and thermal heating constraints. The piezoresistors are n-doped polycrystalline silicon deposited by chemical

vapor deposition and are connected to aluminum contact pads on the outer base of the force sensor via aluminum traces. The sensors are placed at the base of the structure to maximize the strain imposed upon the resistors.

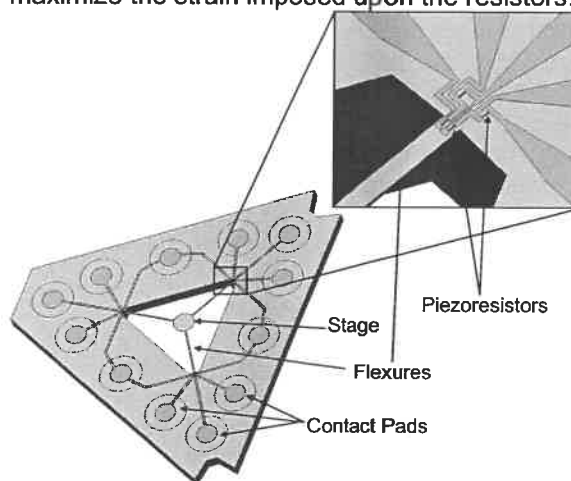


FIGURE 2. Sensor with polysilicon piezoresistors

The flexures in the force sensor are fabricated using deep reactive ion etching (DRIE) of a silicon-on-insulator (SOI) wafer. The flexures are connected to a 1 mm diameter central stage which is meant to contact the cells during testing. The flexures' lengths are limited to 2.5 mm due to device footprint constraints. The flexures' widths and thicknesses are set to meet both the range and resolution functional requirements using the noise flow model. A 70 μ m beam width and 10 μ m beam thickness were chosen.

HexFlex Motion Stage

The force sensor is mounted upon the HexFlex nanopositioner as seen in Figure 3.

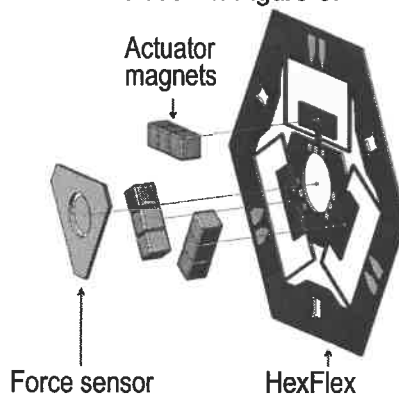


FIGURE 3. Exploded view of HexFlex-sensor system

The HexFlex consists of (1) six bent beams with integrated piezoresistive sensors, (2) three sets of magnets, and (3) a flex circuit with mm-scale coils. The bent beams enable the stage to move in 6 DOF and the piezoresistors enable six-axis displacement sensing. The magnet arrays and flex circuit are used to create six Lorentz force actuators that control the motion of the nanopositioner. The force sensor is placed onto the central stage of the HexFlex. The bond pads on the force sensor are aligned to pads on the HexFlex so that the signal from the piezoresistors in the force sensor may be ported out of the sensor.

Results

Considering all electrical, thermal, and mechanical noise sources within the noise flow model, the force sensor should exhibit 4 nN resolution and 8 nN accuracy at a 1 kHz bandwidth. The force sensor should also exhibit 90 dB dynamic range at a 1 kHz bandwidth. This dynamic range compares favorably to other multi-axis force sensors that have previously been demonstrated [1,2,4].

The major limiting factor in the accuracy and the resolution of the sensor is flicker noise. This may be seen in Figure 4 where the noise spectral density exhibits a $1/f$ dependence until the frequency passes that of the low pass filter - 1 kHz. This large amount of flicker noise in the force sensor is due to the use of polysilicon piezoresistors which were measured to have a Hooge constant on the order of 10^{-3} .

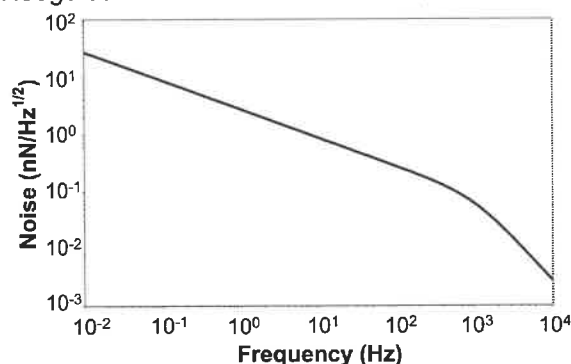


FIGURE 4. Force Noise Spectral Density of 3-axis Force Sensor

Polysilicon tends to have a large amount of flicker noise due to charge carrier depletion at the grain boundaries in polysilicon [10]. The use of other piezoresistor materials such as single crystal silicon [11] or carbon nanotubes [12]

could make it possible to improve accuracy and resolution by at least one order of magnitude.

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DYNAMICAL NON-CONTACT EXHAUSTION PIPE FOR VACUUM COMPATIBLE AIR GUIDED STAGES

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ABSTRACT

The exhaust tubes of high vacuum compatible gas bearing guides should interfere as little as possible with the guide while maintaining a flexible connection and a highly effective exhaustion rate. Within this paper, a novel exhaust system is described that implements these requirements. The major achievement is the realization of a non-contact sealing between two exhaust tubes slidable into one another. This setup is characterized at static and dynamic conditions. The number of seal stages and the clearance height are identified as major impact factors on the leakage rate of the setup. It is concluded that the investigated approach is very suitable for the guides, allowing for an efficient exhaust and avoiding mechanical disturbing forces.

INTRODUCTION

The outstanding qualities of linear gas bearing guides like positioning and guiding accuracy can be used for many positioning tasks in a vacuum environment. The common method to keep the vacuum quality is to exhaust the supplied gas after leaving the bearing area and before reaching the vacuum chamber [1]. Therefore, the gas bearing pads have to be enclosed by exhaust grooves which are separated by sealing areas. The exhaust grooves are connected to atmospheric pressure or to vacuum pumps [2].

An efficient exhaustion rate can be realized by using exhaust tubes with a high conductance which requires that the length of the tube is made as short as possible and the cross-section as large as possible. In addition, it is essential to exhaust the gas from the exhaust grooves without generating disturbing forces or moments to maintain

the precision of the guide. Thus, it is advisable to use a contact-free connection type between the vacuum pumps and the linear guide.

A known way to realize that approach is the integration of the exhaust pipes into the stationary part of the guide [3]. A disadvantageous aspect of this approach is the limited potential of increasing the conductance of the integrated exhaust pipes and the effect of a rising gas leakage with an increasing motion range.

Within this paper, a novel approach is presented for an exhaust system using a contact-free connection with a high exhaust efficiency and a gas leakage that is almost independent from the motion range. The objective of the paper is the characterization of the static and dynamic behavior of the developed exhaust system. Impact factors on the leakage rate of the sealing are analyzed.

SETUP

The contact-free connection between the exhaust groove and the vacuum pump was accomplished by two exhaust tubes which are slidable into one another, separated by a non-contact seal unit. A clearance seal located at the vacuum-sided end of the ringlike gap between the tubes realizes the performance of the seal unit, see fig. 1.

The investigations were performed on a exhaust system separated from the gas bearing guide (see fig. 2) in order to achieve a setup with reduced complexity and to enable the characterization of the system for a widespread set of input parameters.

The sliding tube was made of ceramics whereby

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the high specific stiffness minimizes the gravitational deflection and allows for the realization of a small clearance height of the seal gap. This tube was mounted on a linear rolling guide that was fixed inside a vacuum chamber and could slide through the seal unit into a fixed metal exhaust tube that was connected to a vacuum pump. Via a PTFE hose, nitrogen could be supplied to the sliding tube.

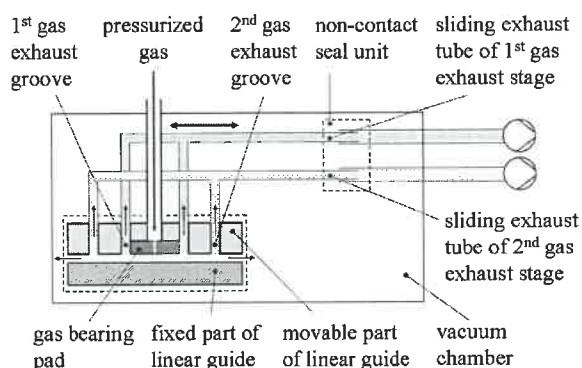
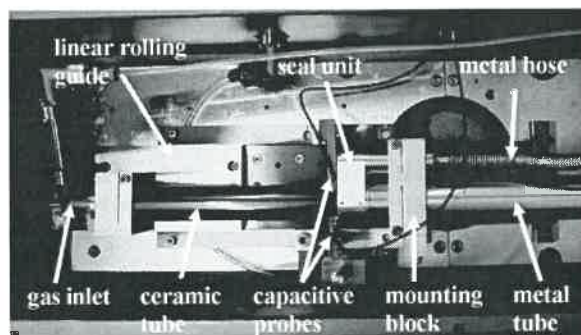
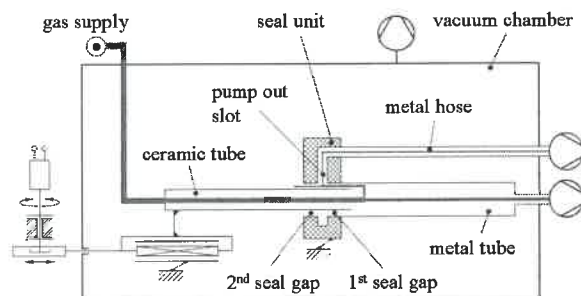


FIGURE 1. Schematic of a vacuum compatible gas bearing guide with two-stage exhaust system.



(a) Setup mounted inside the vacuum chamber.



(b) Schematic of the setup.

FIGURE 2. Design setup 1.

The wide range of the mass flow rate and the pressure respectively occurring in the exhaust tubes of a gas bearing guide was the reason for using two different setups. Setup 1 was used to simulate the higher mass flow rate of the first exhaust stage. Therefore, the seal unit consisted of two sealing surfaces with a connection to an additional vacuum pump in between that could be used to discharge the major part of the gas escaping from the first seal gap (see fig. 2). When the smaller amount of gas of the second exhaust stage had to be discharged, the metal hose and the additional vacuum pump were removed for setup 2.

The first step of the preliminary experimental settings was the alignment of the ceramic tube with the axis of movement of the linear rolling guide. Afterwards, the seal unit was adjusted and fixed by an adjustment bonding process in order to achieve a non-contact movement. During the investigations three different seal units were analyzed. Seal units 1 and 2 were configured to realize a mean gap height of 50 μm and seal unit 3 of 35 μm .

EXPERIMENT

The described setup was used to perform stationary and dynamical tests. Six different mass flow rates for each setup simulated the range of pressure within the exhaust pipe of a real linear gas bearing guide (see tab. 1).

TABLE 1. Mass flow rate.

	mass flow rate / kg s^{-1}		
	$\dot{m}_1$	$\dot{m}_2$	$\dot{m}_3$
setup 1	$3 \cdot 10^{-7}$	$1 \cdot 10^{-6}$	$4 \cdot 10^{-6}$
setup 2	$3 \cdot 10^{-8}$	$7 \cdot 10^{-8}$	$2 \cdot 10^{-7}$
	$\dot{m}_4$	$\dot{m}_5$	$\dot{m}_6$
setup 1	$9 \cdot 10^{-6}$	$2 \cdot 10^{-5}$	$6 \cdot 10^{-5}$
setup 2	$3 \cdot 10^{-7}$	$6 \cdot 10^{-7}$	$1 \cdot 10^{-6}$

For the stationary tests, the length of the traverse path (190 mm) was divided equally by five measuring points. The ceramic tube was stopped at these points and the pressure inside the vacuum chamber was recorded after reaching a quasi-stationary status.

During the dynamical tests the tube was moved along the traverse path with five different maximum velocities and with a different number of cycles (see tab. 2). The pressure inside the cham-

ber was recorded continuously. Every velocity was applied using all six mass flow rates.

TABLE 2. Max. velocities and number of cycles of dynamical tests.

	v_1	v_2	v_3	v_4	v_5
velocity / mm/s	8	20	40	80	160
cycles	10	10	20	40	80

This procedure was performed using the three seal units and both setups. The seal was not replaced during the modification from setup 1 to setup 2.

With the help of the measured pressure difference Δp inside of the vacuum chamber, the leakage rate q_L was calculated using the pumping speed of the vacuum pump S_p .

$$q_L = S_p \cdot \Delta p \quad (1)$$

RESULTS

Stationary tests

The results for the stationary tests for setup 1 are shown in fig. 3a and for setup 2 in fig. 3b. The six groups of measuring points represent the six different mass flow rates that were supplied. Each group consists of five points which represent the five different positions of the ceramic tube during the tests.

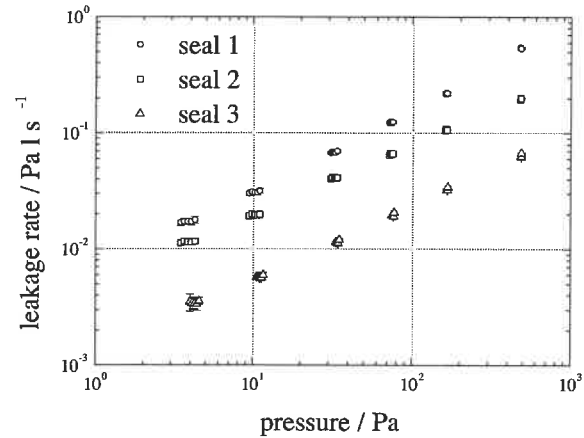
The leakage rate rises almost linearly with the supplied mass flow and the calculated pressure in front of the seal unit, respectively. It is slightly influenced by changing the position of the ceramic tube as well due to the variation of the flow resistant between the two tubes. With increasing retraction depth of the ceramic tube, the leakage rate decreases.

The pressure p_p at the vacuum pump that is connected to the metal tube can be calculated by

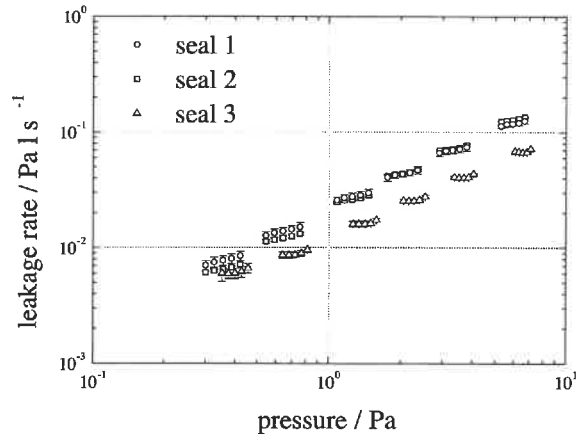
$$p_p = \frac{\dot{m} R_s T}{S_p} \quad (2)$$

Hereby R_s is the specific gas constant, T the temperature of the vacuum chamber and S_p the pumping speed of the vacuum pump. With the known geometrical dimensions and p_p , the relevant pressure p_s in front of the seal can be calculated using the common relationships of the flow

characteristics through pipes. The fluid dynamics have to be considered during these calculations because typically all different kind of flow regimes can occur (molecular, intermediate and viscous flow).



(a) setup 1.



(b) setup 2.

FIGURE 3. Stationary leakage rate with increasing pressure in front of sealing.

Inside the seal gap, a molecular flow exists due to the low pressure and low clearance height. The analytical relationship between the leakage rate q_L and the number of seal gaps n and the gap height h_g is an exponential one:

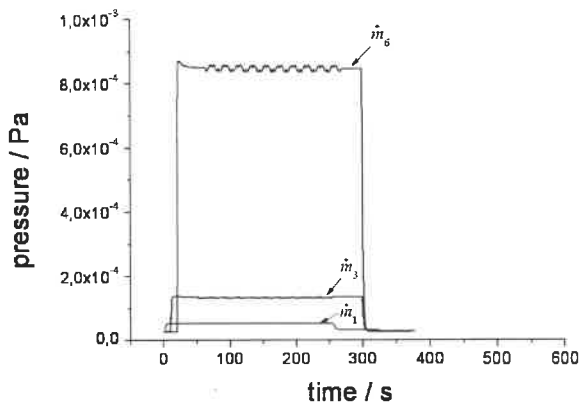
$$q_L \propto h_g^{2n} \quad (3)$$

Consequently, the measured leakage rate of setup 1 is approximately one order of magnitude below the leakage rate of setup 2 at comparable input conditions. The strong impact of the seal

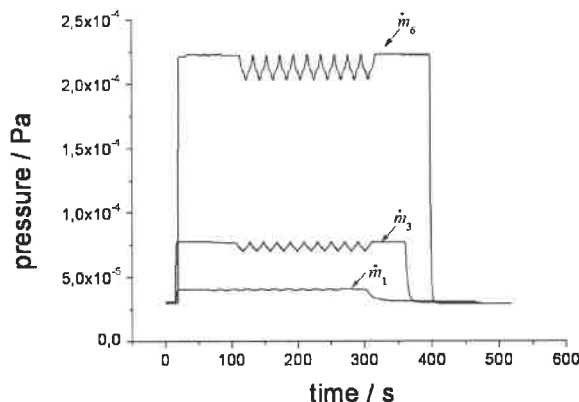
gap height on the leakage rate is particularly clarified at setup 1. The reduction of the gap height from seal unit 2 ($50\text{ }\mu\text{m}$) to seal unit 3 ($35\text{ }\mu\text{m}$) causes a decrease of the leakage rates of about 70 % (theoretical 76%).

Dynamical tests

The pressure inside the vacuum chamber during the dynamical tests is plotted in fig. 4. The figures show exemplarily the pressure of both setups for seal 1, velocity v_2 and mass flow rates $\dot{m}_1$, $\dot{m}_3$ and $\dot{m}_6$.



(a) setup 1.



(b) setup 2.

FIGURE 4. Pressure inside the vacuum chamber during dynamical tests (seal unit 1, v_2).

The run of the curves has to be read as follows: The first rise in pressure is caused by opening the gas inlet. At this point of time the tube stands still. During the movement of the tube, the pressure is oscillating. After passing the determined cycles, the movement of the tube and the oscillating of the pressure stop. The steep fall of pressure shows the closure of the gas inlet.

The change in pressure caused by the movement of the ceramic tube lies within the same range as the difference of the pressure while moving from the first to the fifth measuring point during the stationary tests.

Regarding setup 1 (fig. 4a), the change of the pressure caused by moving the tube is almost negligible compared to the total pressure rise caused by feeding the gas. On the other hand, the change of the pressure caused by the movement of the tube equals approx. 10% of the total pressure rise in setup 2. The origin of this behavior is the additional air exhaust stage used in setup 1.

If the results of the stationary and the dynamical tests are compared, a relation between the speed of the ceramic tube and the leakage rate could be observed solely for setup 2 and low mass flow rates. In this case, a slight increase of the leakage rate above the static values was detected. For all other dynamical experiments, a correlation between speed and leakage rate could not be determined.

CONCLUSIONS

The developed exhaust system shows tolerable leakage rates for the application in a high vacuum compatible gas bearing guide. During all experimental tests, a high vacuum level in the order of 10^{-4} Pa has been maintained within the vacuum chamber. The gap height of the applied clearance seal has to be as small as possible and the number of seal stages should be well chosen to minimize the additional leakage of the seal unit. A significant influence on the leakage rate caused by a dynamical movement of the sliding tube could not be detected.

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THE MEASUREMENT OF MESO-SCALE TORSION SPRINGS

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INTRODUCTION

The characterization of meso-scale torsion springs is an important activity for applications at Sandia National Laboratories relying on the miniaturization of mechanical mechanisms. Designers continue to push the design limits for springs, and are faced with significant uncertainty in the quantification of the static, dynamic and fatigue behavior of small torsion springs. While static stiffness is the primary parameter specified in designs, providing in-situ measurements of dynamic stiffness, fatigue life and operational torque loads would give designers information necessary for further optimization and improved reliability. Because commercial systems do not exist that meet the requirements for range, accuracy or dynamic response, this paper will describe work to develop a measurement system for the characterization of meso-scale torsion springs.

MESO-SCALE TORSION SPRINGS

Figure 1 shows an example of a spring targeted by the new measurement system. They are produced through traditional coil winding processes and typically consist of drawn Elgiloy, Hastelloy, Inconel or stainless steel wire. Coil diameters and lengths are typically 4mm or less with wire diameters spanning from 50 to 500 μ m. Mounting methods vary with the design application, so end geometries may have curved or straight tangs with radii of curvature and bend angles in any orientation. Operational torque loads are typically 0.5 to 2.0mN·m, with initial preloads in the range of 0.5 to 1.5mN·m. Normal mechanism motions rely on 20-30° of additional spring rotation beyond their initial preloads at cycle rates up to 100Hz.

MEASUREMENT SYSTEM REQUIREMENTS

While systems for measuring torsion springs exist¹, no commercial equipment meets the requirements for meso-scale torsion springs. Fundamental shortcomings of commercial systems are their inability to measure dynamic

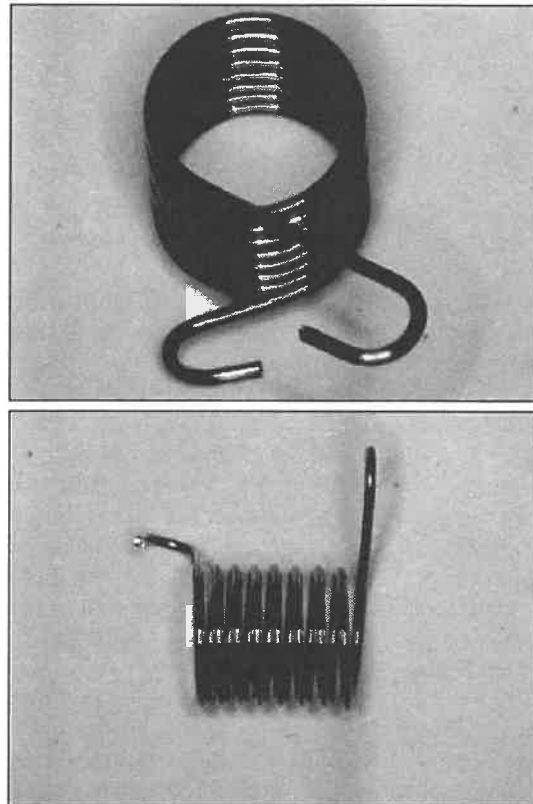


FIGURE 1. A representative meso-scale torsion spring. The wire diameter is 0.175mm, the spring length is 3.8mm, the coil diameter is 2.5mm, and the wire material is Elgiloy.

spring motions or to quantify torque below 10mN·m^{2,3,4}. Additional problems with measurement accuracy, with traceability and with system reliability have been observed with customized equipment delivered to SNL. The presented work details a system developed to address each of these weaknesses. The proposed system has been designed to measure in-situ torque ranging from 0.1 to 10mN·m, at frequencies up to 100Hz and with dynamic rotations of $\pm 15^\circ$. Such requirements enable the measurement of both static and dynamic spring rates. As a result, the effects of

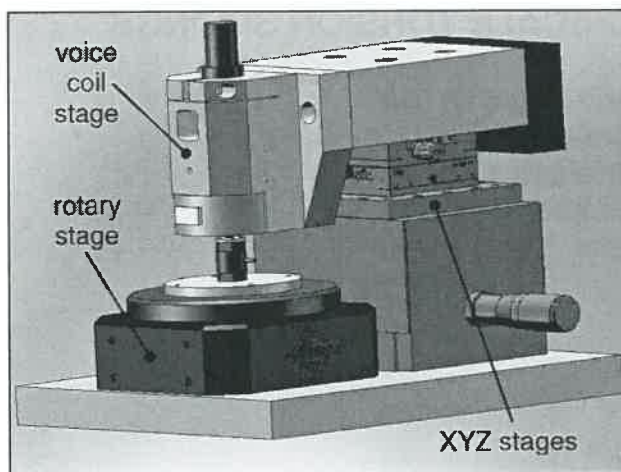


FIGURE 2. The prototype system for the measurement of meso-scale torsion springs.

resonance, cyclic loading, fatigue, end effector geometry and even defects on operational performance will be quantifiable.

SYSTEM DESIGN

The prototype measurement system is presented in Figure 2. The system was designed to be simple, and yet to provide the functionality necessary to meet the design requirements. Critical elements of the design are its torque sensor, actuation mechanism, interfaces for spring mounting, and axes of motion for system alignment.

Torque Sensor

One of the most critical system components is the torque sensor as its performance provides the primary foundation for the accuracy of torque measurements. After a detailed search, a Kistler 9329A torque cell and 5015A charge meter were selected for the system. Several different techniques exist for measuring torque⁵, but only piezoelectric and piezoresistive (i.e. strain gage) sensor technologies exist in commercial forms useful for meso-scale springs. Traditional strain gages provide common solutions for torque measurements, but their inadequate measurement range and their low frequency response drove the selection of a PZT sensor with the desired range, resolution and bandwidth.

While the 9329A provides a maximum measurement range over $\pm 1\text{N}\cdot\text{m}$ and a threshold of $0.03\text{mN}\cdot\text{m}$, the 5015A charge meter provides ranges adjustable down to $\pm 1\text{mN}\cdot\text{m}$. The natural

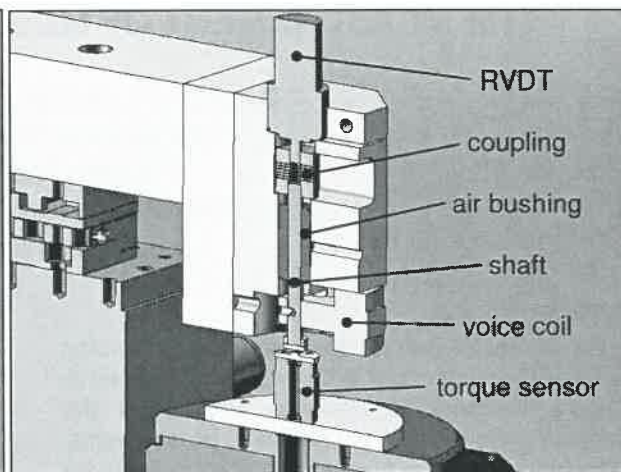


FIGURE 3. A cross-sectional view of the prototype system identifying critical elements of the actuation mechanism.

frequency of the sensor is greater than 50kHz , and therefore provides no problems for measuring 100Hz motion. The $\pm 0.03\text{mV/sec}$ drift rate of the charge amplifier, however, does restrict measurements to the "quasi-static" regime. Real time data from the torque sensor will be captured by a PC using a DAQ card and LabView software.

Actuation

Actuation presented another important design challenge due to the required response time and range of motion for dynamic testing. Figure 3 shows a cross-section view of the actuation elements in the system. Large spring rotations, and hence static preloads, are created by an Aerotech ADRS-150 rotary stage which provides 360° of continuous motion with sub $10\text{-}\mu\text{sec}$ resolution. The Kistler torque sensor is mounted to the rotary stage, allowing direct measurement of spring torques without contributing to the dynamic mass. Control of the stage is performed using the standalone Aerotech NViewMMI software on the same PC used for acquiring torque data. An important feature of the ADRS-150 is its use of a mechanical brake which provides a more stable pre-load to the springs under dynamic test loads.

Higher frequency spring rotations are created using a rotary voice coil from BEI-Kimco. A RA29-11-002A is currently specified for the system, although a custom device is still under consideration. The RA29-11-002A has a $\pm 16^\circ$ stroke, $226\text{mN}\cdot\text{m}$ of peak operating torque and a

mechanical time constant of 23msec. Although PZT actuation was considered due to its excellent dynamic response, inadequate travel ranges prompted the decision to use voice coil actuation. Control of the voice coil actuator is initially planned to be performed open loop using a function generator and a current amplifier.

Since the rotor of the voice coil requires a bearing to fix its pivot point and its planar location within the stator, a New Way 0.25in air bushing and shaft were selected for this function. The air bushing provides very low friction and high stiffness which are ideal for high frequency motion. The bushing was also incorporated directly into the housing sub-assembly, simplifying assembly and maintaining a compact, low mass design. Due to the small length of the shaft required by the system components, the small torques and moments required to test meso-scale torsion springs, and the effort to minimize shaft mass, a single bushing was deemed acceptable for the actuator. Coupling of the shaft to the voice coil is achieved using a set screw.

Shaft rotation will be measured using a Schaevitz R30D rotary variable differential transformer (RVDT). It has a linear measurement range of $\pm 30^\circ$ and a 3dB bandwidth of 500Hz. The sensor requires a 15V DC input, and produces approximately 0.32mN·m of resistance torque. The sensor is coupled to the actuator shaft using a flex coupling available from the vendor. The RVDT signal will be initially monitored open loop using LabView, but will eventually enable closed loop control of the voice coil actuator.

Each component in the actuation sub-assembly was selected or designed to maximize the bandwidth of the moving mass. As a result, mass moments of inertia, J_o , were calculated and subsequently minimized to predict and optimize the performance of the actuator. Table 1 lists values for the moving components in the actuation assembly, and provides the corresponding torque required for $\pm 12^\circ$ motion at 50Hz. Torque values were obtained assuming forced harmonic motion with no damping. The governing equation of motion is therefore

$$J_o \cdot \ddot{Q} + k_Q \cdot Q = T_o \cdot \cos(w \cdot t)$$

where J_o is the component mass moment of inertia, Q is the angular position, k_Q is the angular stiffness, T_o is the driving torque amplitude, w is the angular velocity and t is the time. The amplitude of the necessary harmonic drive torque can be calculated from the relationship

$$T_o = Q_{max} (k_Q - J_o \cdot w^2)$$

where Q_{max} is the amplitude of angular motion. Since, the RA29-11-002A actuator provides 226mN·m of peak torque, Table 1 reveals that the voice coil is capable of driving every component in the assembly except for its own rotor. Based on its specifications, it should be able to achieve 43.5Hz motion with maximum amplitude of $\pm 12.7^\circ$, assuming no other attached components. Therefore, work continues with BEI-Kimco to develop a custom voice coil which can meet the system design requirements.

TABLE 1. Mass moments of inertia for the moving elements of the voice coil assembly. Torque values are those required to produce $\pm 12^\circ$ motion at 50Hz.

Component	J_o , kg·mm ²	T_o , mN·m
torsion spring	0.0006	0.055
spring mount	0.0115	0.237
shaft	0.0776	1.604
RVDT	0.0155	0.320
flex coupling	0.630	13.02
voice coil rotor	13.6	281

Spring Interfaces

The mounting of small springs also presents a design challenge due to difficulties in handling and the influence of end loading conditions. Custom mounts will be designed for each spring design due to the wide range of spring dimensions and end geometries. These mounts will be assembled directly to the torque sensor and the voice coil actuated shaft. They will function to mimic application loading conditions and therefore use center mandrels and either pins or slots in the proper tang location. Figure 2 shows that a Newport MVN120 vertical stage is used to drive the voice coil subassembly clear of the spring mounts to facilitate spring loading. The MVN120 provides 20mm of travel, 1 μ m resolution and a maximum load capacity of 400N. It was placed under the two horizontal stages to reduce moment loads and to therefore

improve structural stability. While currently very simple, the system could be easily adapted to incorporate a motor driven stage and a three-axis load cell. Automatic spring loading could then be implemented to facilitate small spring handling and to prevent part damage.

System Alignment

On top of the vertical stage is mounted a stacked set of Newport TSX-1D dovetail linear slides. The slides provide the planar motion necessary to align the rotational axes of the Aerotech ADRS-150 and the voice coil to insure proper rotational loading of the torsion springs. Dovetail slides were chosen for their simplicity and to reduce cost. While motorized stages could be used, the planar motion is only necessary during system assembly, after which the stages can be locked to maintain alignment. Because spring loading is insensitive to small position errors, the 4 μ m resolution of the slides is adequate for accurate torque measurements. The torque cell and its spring mount will be initially aligned to the rotary stage using an indicator mounted onto the system base plate. The voice coil will then be aligned to the rotary stage by mounting the indicator onto the table of the rotary stage and indicating to the spring mount attached to the rotating shaft.

Although the TSX-1D slides provide 25mm of travel, an effort was made to minimize stage excursions by nominally aligning the axes of the voice coil and the rotation stage when the stages are at their center of travel. The voice coil actuator was also counterbalanced to stabilize the system and to improve the rejection of structural disturbances. The total mass above the lower dovetail slide was 65N, well below the 111N limit for the stage. Structural resonances of the system were examined through FEA analyses to insure that no resonances would be excited during spring testing. The lowest resonant frequency of the system was found to be at 842Hz and corresponds to cantilever bending of the voice coil actuation assembly on top of the motion stages. The next mode is at 907Hz and corresponds to rotation of the voice coil and counterbalance about the vertical axis of the motion stages. While the rotational motion during spring testing would most likely excite this mode, its frequency being an order of magnitude larger than the desired test frequencies suggests a minimal influence.

CONCLUSIONS AND FUTURE WORK

The design of a prototype system for the dynamic torque testing of meso-scale torsion springs has been completed. Future work will focus on system implementation and will involve procuring, fabricating and assembling hardware; implementing real time computer feedback and control; developing calibration procedures; quantifying system performance and measurement uncertainties; and testing of actual springs.

ACKNOWLEDGEMENTS

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DESIGN AND CONTROL OF A NANO-PRECISION MULTI-AXIS VERTICAL MASK ALIGNER

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INTRODUCTION

The semiconductor industry is a higher value-added business. In addition the semiconductor manufacturing techniques are required high technological process. Now the techniques come at dozens nanometer units. Among other semiconductor manufacturing equipments, mask aligner is one of the most important equipment. It makes mask to align on the wafer surface precisely. It must have three degree of freedom, one translation and two rotational motions. In addition the precision of mask aligner is dozens nanometer units. In these days, there are many precision mask aligners. The majority of them have a horizontal framework type. It means that mask and wafer are parallel with ground. Those types have the merit of simple construction and stabilities. But recently they reach uppermost limit. That is the gravity. As time went on, the mask and wafer were getting bigger. In addition their precision rates were getting more important. But gravity makes mask to warp. Consequently in the future, it will be nearly impossible to manufacturing ultra precision (nanometer-unit) semiconductor

VERTICAL MASK ALIGNER DESIGN

Lately, because of these problems, their framework types are changed to perpendicular to ground. Therefore the wafer and mask are vertical to ground, too. It overcomes a handicap of the horizontal type. Because mask surface and wafer parallel to gravity force. Therefore, there is not curved phenomenon at the mask surface. It is possible to make bigger mask and wafer with a nanometer-precision. Therefore, in these days many semiconductor manufacturing corporations have researched this type equipment. But, up to present, they all work in the lab, and cannot be deployed on a commercial scale. This study covers vertical type mask aligner stage system. Especially, it analyzes 3-DOF guide hinge. It is called rotational symmetric leaf spring type hinge. For using this guide hinge, it needs to analysis of 6-DOF stiffness of the guide hinge. And it is driven

by PZT Actuator. The PZT has very small moving range. Therefore it needs motion amplification mechanism. The motion amplification mechanism consists of 12-flexure hinges. Total system uses three guide hinges and three PZT amplification mechanisms and one moving plate. The moving plate is where the make is fixed on. Total system consists of 39-flexure hinges and 31-rigid bodies, and it carries out optimal design process totally. Using optimal design value, it was simulated by FEM tool. To prove validity of the analysis of 6-DOF stiffness, it is compared an analytic model with a FEM model. The error is below the 10%.

Amplification Mechanism

The vertical mask aligner stage consists of three piezoelectric actuators. PZT actuator has short transport distance. Therefore, it will need amplification equipment. The displacement generated by a piezoelectric actuator is 50 μ m. Amplified by the amplification equipment are 200 micro-meters displacement amplification.

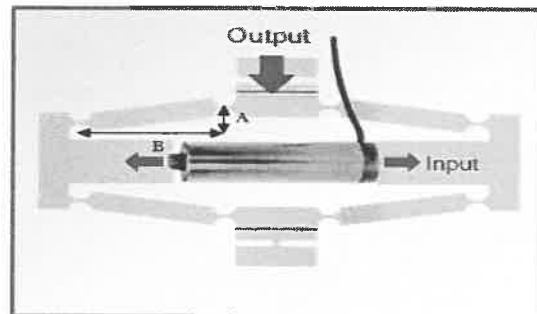


FIGURE 1. Proposed AMP mechanism

Guide Mechanism

The mask aligner needs three degree of freedom, one translation and two rotational motions. Therefore, the guide hinge needs to three degree of freedom, too. The rotationally symmetric hinge [1] has 3-DOF, one translation and two rotational motions. It connects with the shaft coupler was used to. However, it is also easy to use as a mask aligner guide.



FIGURE 2. Proposed Guide Hinge

CONCEPT DESIGN

Total system uses three guide hinges and three PZT amplification mechanisms and one moving plate. The moving plate is where the make is fixed on. Total system consists of 39-flexure hinges and 31-rigid bodies. The system is compared FEM simulation tool. The error is below the 10%.

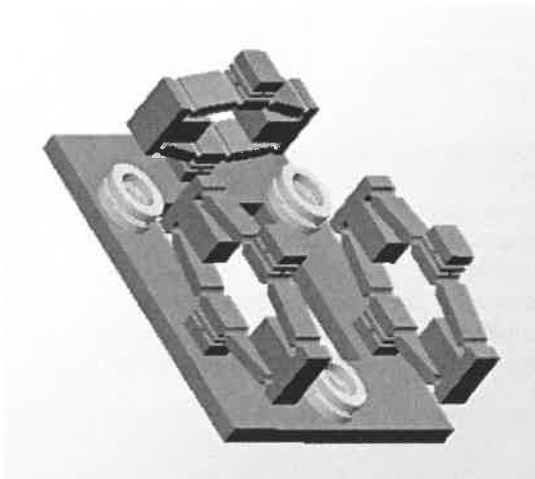


FIGURE 3. Proposed Mask Aligner Stage

FEM SIMULATION RESULTS

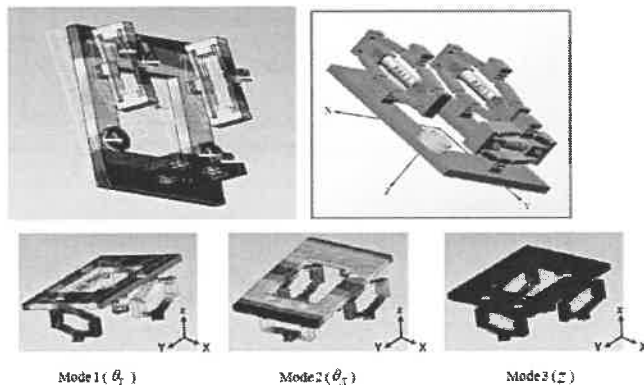


FIGURE 4. FEM Simulations.

1. Static result

	PZT 1	PZT 2	PZT3
Force	10 N	100 N	100 N
direction	Modeling	FEM	Error
Z direction	$-2.09 \times 10^{-5} m$	$-1.92 \times 10^{-5} m$	-8.85%
θ_x direction	$1.42 \times 10^{-4} rad$	$1.44 \times 10^{-4} rad$	1.40%
θ_y direction	0 radian	0 radian	0 %

2. Dynamic results

Mode	Modeling	FEM	Error
1 mode	161 Hz	155 Hz	4.3%
2 mode	170 Hz	169 Hz	0.5%
3 mode	171 Hz	170 Hz	0.5 %

CONTROL OF THE STAGE

The proposed stage is parallel mechanism in which three PZT actuators are coupled. Therefore, the kinematic relationship between input voltage of each actuator and motion of moving part must be examined. Since three sensors are also coupled, the same operation between sensor output and motion of moving part must be conducted [2].

$$F = TV_{in} \quad (1)$$

$$X = T^*V_{out} \quad (2)$$

In above equations F is the force vector by PZT actuator, X is the displacement vector of moving part expressed by rectangular coordinate, T and T* is the transpose matrix containing the kinematics of actuators and sensors, V_{in} is actuator input voltage vector and V_{out} is sensor output voltage vector.

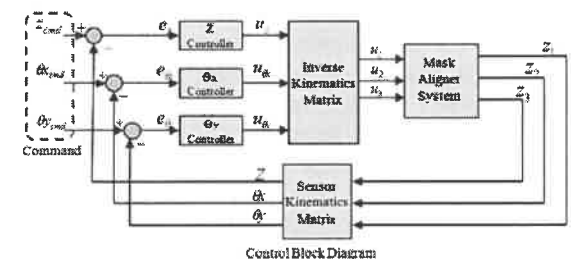
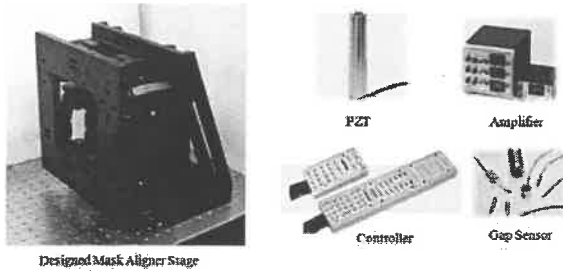


FIGURE 5. Control Block Diagram.

EXPERIMENTAL VERIFICATION



Designed Mask Aligner Stage

FIGURE 6. Developed Vertical Mask Aligner



FIGURE 7. Hardware components of the control system.

The experiment result as follows. The stroke of translation movement is 190 micrometer, and the stroke of rotation movement is 0.5mrad. And in-position stability is ± 2 nm, ± 30 nrad, and resolution is 6 nm, 80nrad. The settling time is below the 50msec.

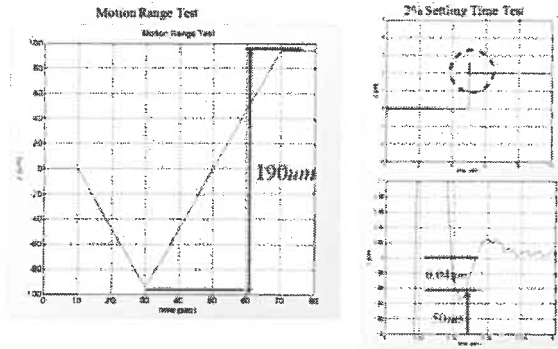
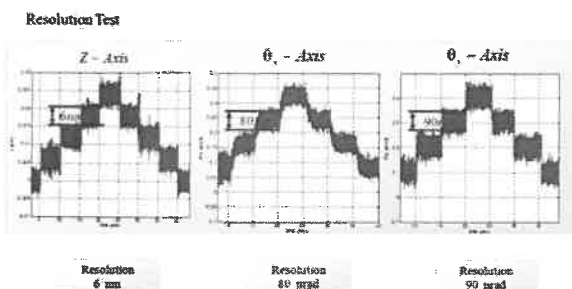
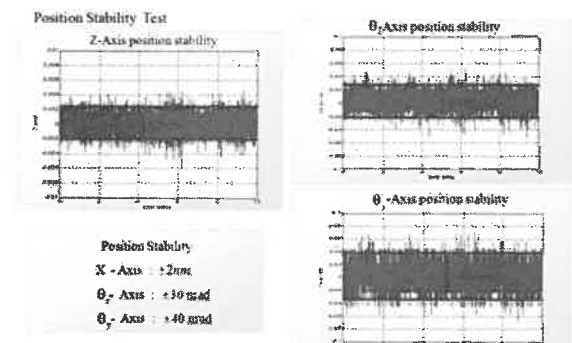


FIGURE 8. Experiment Results.

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DESIGN AND TESTING OF A STEP COMPENSATED BI-CONIC HYDROSTATIC SPINDLE

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ABSTRACT

In high precision machining and grinding applications, high-stiffness, high-load capacity workhead spindles and rotary tables with nanometer-level error motions are becoming more necessary. The combination of reasonable load capacities and 10,000 RPM maximum speed make the Professional Instruments Company model 4R BLOCK-HEAD[®] air bearing spindle a first choice for many high-accuracy applications. However, there are applications requiring more load capacity in the same package size, making high pressure oil hydrostatic a logical candidate. The present work describes the design and testing of a step compensated oil hydrostatic spindle consisting of two conical bearing surfaces (bi-conic) with high load capacity—ideal for the rigorous demands of precision machining and grinding. The most compelling argument for this oil hydrostatic spindle is the combination of nanometer-level error motions and near indestructibility.

INTRODUCTION

Hydrostatic spindles featuring conical surfaces have been shown to be very effective in providing precision rotation and high load capacity in a compact, low profile package [1]. The present bi-conic oil hydrostatic spindle is unique in that it uses step compensation rather than restrictor compensation with pockets. This results in high load capacity and stiffness since the average pressure of the bearing film is a higher percentage of supply pressure.

Because of the excellent performance of the 4R BLOCK-HEAD[®] air bearing spindle, we use its specification as a standard of comparison for the characteristics of this new design. The 4R bi-conic spindle features a mounting interface identical to the 4R BLOCK-HEAD[®]. Furthermore, with modifications to the step and the bearing

clearance, the 4R bi-conic bearing can be designed to operate with air.

It is impractical to design an air bearing to use air pressure above 150 PSI, while pressurized oil hydrostatic systems supplying greater than 600 PSI are common. When supplied with high pressure oil, the 4R bi-conic spindle offers significant improvements in load capacity and provides similar performance in error motion and stiffness as compared to the BLOCK-HEAD[®]. In addition, the beneficial squeeze film effects help to prevent damage to the bearing surfaces during a crash.

DESIGN

The motivation for step compensated bi-conic spindle development at Professional Instruments Company is derived from several potential advantages of a bi-conic design including:

- Increased bearing efficiency (load capacity for a given volume)
- Increased rotor stiffness (enabling the use of higher bearing pressures)
- Lower thermal sensitivity
- Fewer components
- Better balance of radial, axial and tilt load capacities

Figure 1 illustrates the design differences between the three-piece BLOCK-HEAD[®] rotor and the two-piece bi-conic rotor. The BLOCK-HEAD[®] uses separate bearing surfaces to provide axial, radial and tilt load capacity, while the bi-conic design uses the same surface to provide axial, radial, and tilt capacity.

The axial and radial load capacity of the spindle is directly related to the projected surface area in the axial and radial directions. Table 1 shows that while the bi-conic design has approximately 5% less axial area, it has approximately 63%

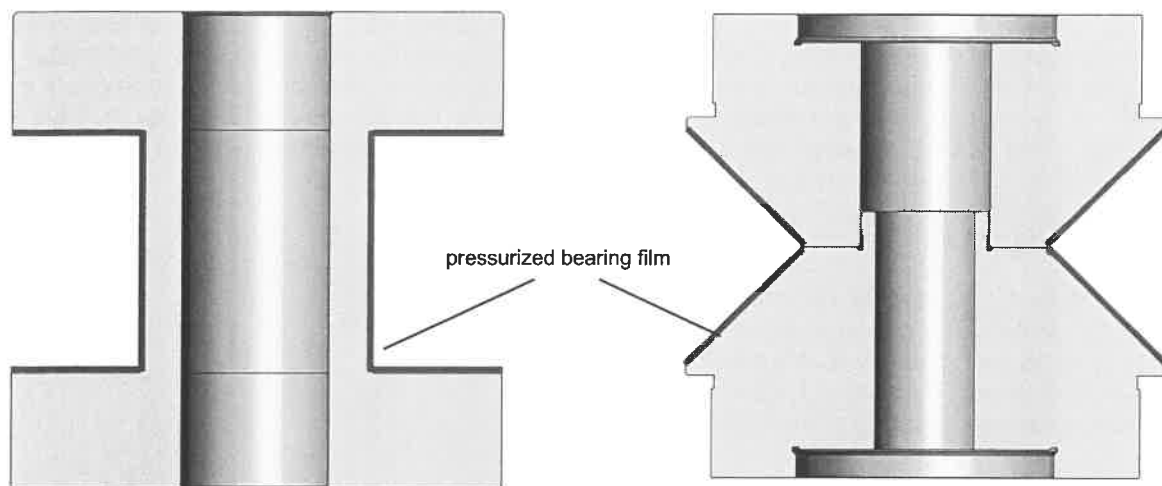


FIGURE 1. Comparison of the rotors of the 4R BLOCK-HEAD® and the 4R bi-conic.

greater radial projected area. Therefore, for a given supply pressure, the bi-conic design will have marginally less axial capacity with significantly more radial capacity.

TABLE 1. Comparison of projected bearing areas.

Model	Axial mm ² (in ²)	Radial mm ² (in ²)
4R BLOCK-HEAD®	6450 (10)	2580 (4)
4R bi-conic	6130 (9.5)	4195 (6.5)

The stiffness of the rotors shown in Figure 1 affects static and dynamic performance. The bi-conic rotor is noticeably stiffer in the axial direction which results in better dynamic performance and enables the use of far higher pressures.

The bi-conic design is less thermally sensitive than a BLOCK-HEAD® spindle due to the geometry of the air film. A temperature difference between the rotor and the stator in either design will result in a decreased clearance but the bi-conic design can tolerate a higher temperature difference. A variation in the design featuring convergence of the cones at a common apex results in a completely thermally insensitive spindle.

The stator and the rotor halves of the bi-conic spindle are manufactured from 416 series stainless steel. This series of stainless steel was chosen because of its corrosion resistance, hardenability, and thermal expansion properties similar to mild steel.

TEST RESULTS

When supplied with oil, the maximum speed of the bi-conic is much lower than the 4R BLOCK-HEAD®, but the 4R bi-conic oil hydrostatic spindle offers significant improvements in load capacity and provides similar performance in error motion and stiffness.

As shown in Table 2, the load capacities of the 4R bi-conic supplied with 600 PSI oil pressure are 3,000 N radial, 9,500 N axial and 320 N-m tilt. The 4R BLOCK-HEAD® capacity at 150 PSI air pressure is 440 N radial, 1,780 N axial and 45 N-m tilt. The real advantage of the bi-conic design is the ability to run at much higher pres

TABLE 2. Comparison of load capacity.

Model	Axial N (lb)	Radial N (lb)	Tilt N-m (lb-in)
4R BLOCK-HEAD® 150 PSI	1780 (400)	440 (100)	45 (400)
4R Bi-conic 600 PSI	9500 (2100)	3000 (675)	320 (2800)

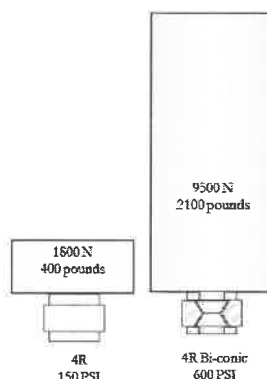


FIGURE 2. Visualization of the axial load capacity of the 4R BLOCK-HEAD® and the 4R bi-conic.

sure without excessive rotor distortion. The axial load capacity is over 5 times greater while the radial and tilt capacities are approximately 7 times higher. The radial capacity of a 4R BLOCK-HEAD® is 1/4 the axial capacity. The 4R bi-conic offers more balance since its radial capacity is approximately 1/3 the axial capacity.

As shown in Table 3, the static stiffness of the oil hydrostatic bi-conic spindle is not significantly increased. This is due primarily to the fact that the oil version of the bi-conic has a larger clearance in order to reduce shearing of the oil film.

TABLE 3. Comparison of static stiffness.

Model	Axial N/ μ m	Radial N/ μ m	Tilt N-m/ μ rad
4R BLOCK-HEAD® 150 PSI	350	120	0.45
4R Bi-conic 600 PSI	350	130	1.1

The improved dynamic stiffness of the oil hydrostatic bi-conic design results in an incredible improvement in the dynamic response as shown Figure 3.

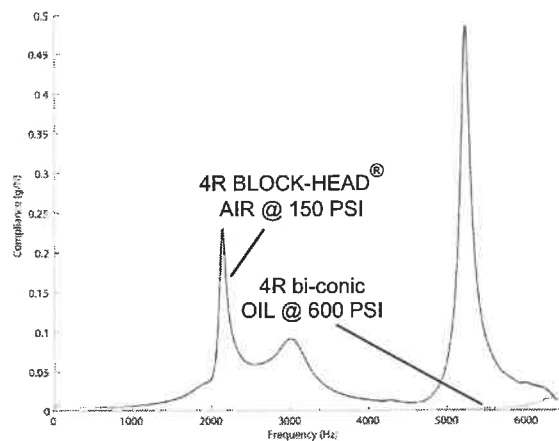


FIGURE 3. The accelerance frequency response function illustrates the superior dynamic response of the oil hydrostatic bi-conic.

For further testing, the bi-conic spindle was motorized with a brushless, frameless motor and amplifier (MCS custom-wound motor and MCS AX-2000 amplifier) and a 1024-count rotary encoder (Renco R22i).

Previous work has described a radial multiprobe error separation procedure using the tooling shown in Figure 5 [2]. Multiprobe separation is used with the sensors located at 0°, 99.844° and

202.5°. The sensor indexing plate is then rotated 90° to measure the orthogonal component of radial error and the multiprobe separation procedure is repeated. For tilt error motion, a similar procedure is followed using the face tooling shown in Figure 6.

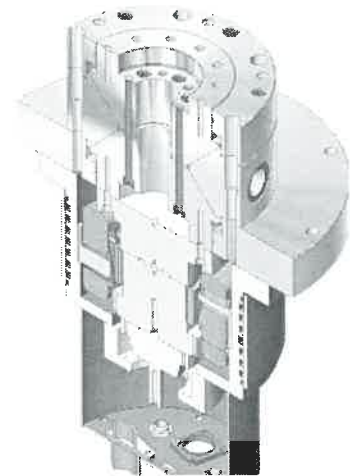


FIGURE 4. Sectioned view of the bi-conic spindle with brushless motor and encoder.

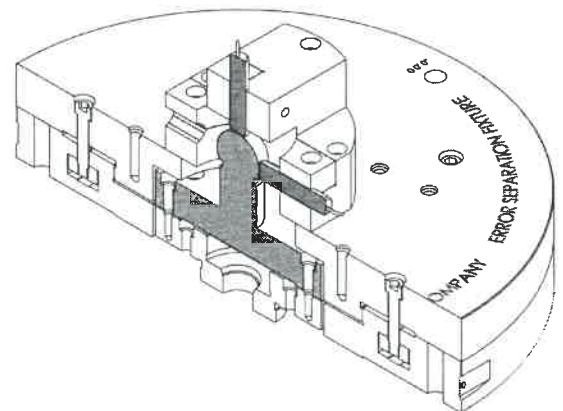


FIGURE 5. Sectioned view of the radial multiprobe error separation tooling.

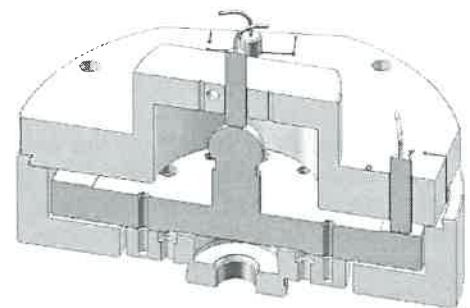


FIGURE 6. Sectioned view of the face multiprobe error separation tooling used to measure tilt error motions.

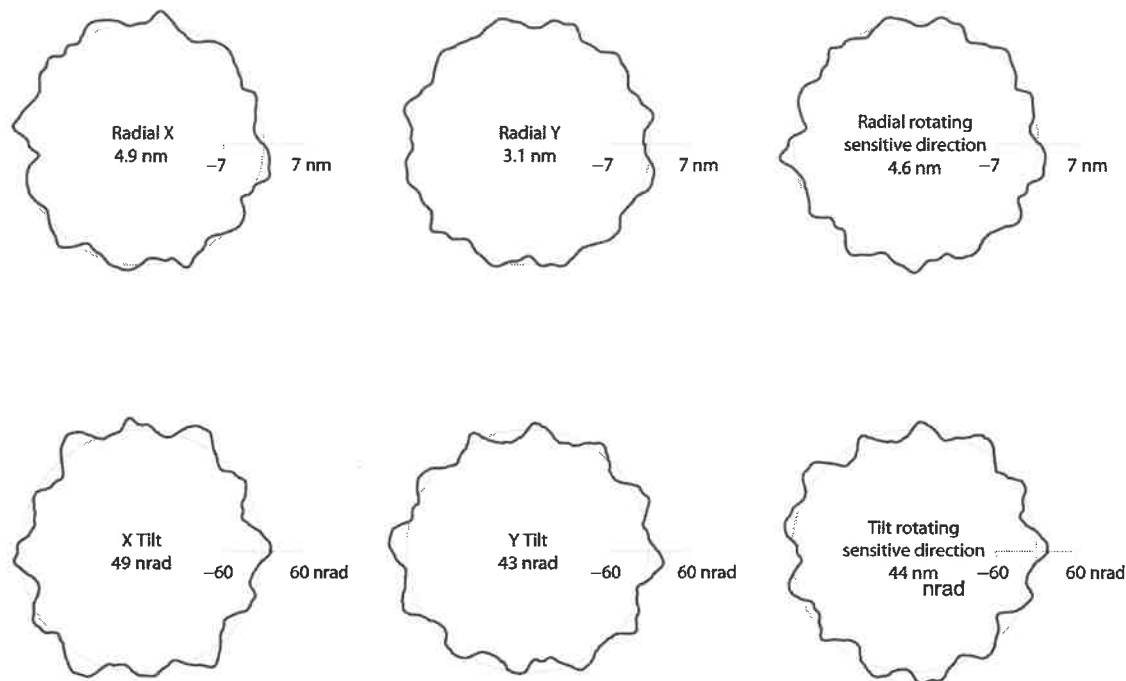


FIGURE 7. Two radial and two tilt degrees-of-freedom at 100 RPM.

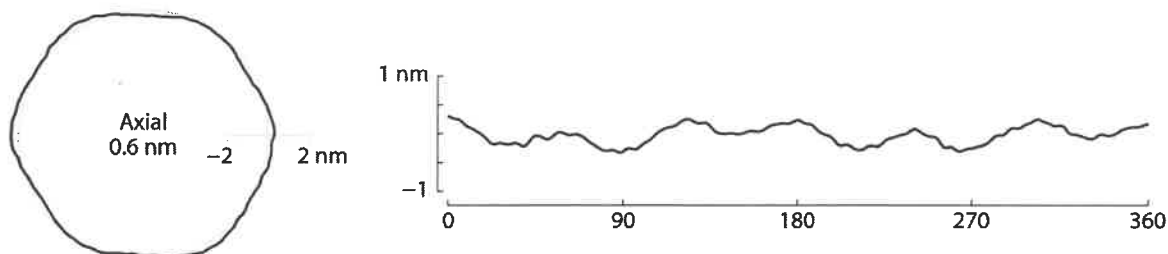


FIGURE 8. Residual axial spindle error motion at 100 RPM. The fundamental axial is 0.1 nm.

The capacitive sensor (Lion Precision C23-C, 0.5 nm/mV) targets an Ø25 mm lapped sphere for radial and an Ø150 mm flat ring for face testing. The amplifier (Lion Precision CPL290) incorporates a 15 kHz first-order, low-pass analog filter with linear phase response. The data acquisition system (Lion Precision SEA V8.3) is triggered by the encoder index pulse.

Synchronous error motions in the five degrees-of-freedom are demonstrated in Figures 7 and 8 at 100 RPM. The radial fixed sensitive direction measurement is 4.9 nm in the X direction and 3.1 nm in the Y direction. The X and Y tilt errors as calculated from the face measurements are 49 nanoradians and 43 nanoradians. The fundamental axial error motion is 0.1 nm and the residual axial error motion is 0.6 nm. The radial, tilt and axial spindle error motion components are defined by the *ASME B89.3.4 Standard* [3].

CONCLUSION

The design considerations and test results of a bi-conic step compensated spindle are described. This spindle is designed to operate with oil but the bi-conic design can be operated with air supply and tighter clearance and a smaller step. Under oil supply, the maximum speed is much lower than an air bearing spindle, but significant improvements are realized in load capacity due to increased supply pressure.

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INSTRUMENT FOR GAUGING LARGE RINGS, CUPS, AND CONES

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INTRODUCTION

Master parts are used in manufacturing facilities to set up comparator-type measurement instruments, which will then be used during production to inspect production parts. The process of "mastering" these instruments is tedious and requires an experienced operator. Also, these master parts are numerous and take up a significant amount of space within a manufacturing facility. In an effort to conserve space, streamline the manufacturing process, and improve operator ergonomics, a new measurement gauge that is capable of fulfilling the measurement needs of the customer, and which does not require individual mastering, is desired.

We are interested in measuring circular parts which are in the shape of shallow rings, cups, and cones; typically inspected using bar gauges. Each bar gauge needs to be set up using a master part, a controlled physical standard of a specific part. On each of these parts there can be numerous critical dimensions that need to be monitored, and transferred from the master part during production. Therefore several bar gauges will need to be mastered on the master part for these dimensions. The simplest of these bar gauges resemble a caliper, except there is no graduated scale for direct measurement (Figure 1); some may contain a micrometer head on one of the jaws with limited range of travel to provide some feedback on the ring's size.

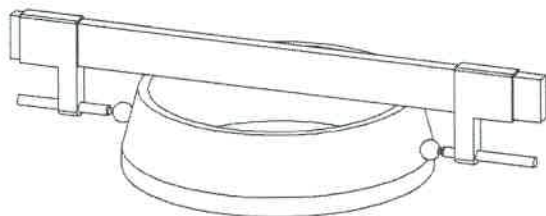


FIGURE 1. Bar gauge on tapered ring (cone), measuring OD.

INSTRUMENT DESIGN

The proposed instrument consists of a mechanism, called the angular encoder caliper,

which resembles a proportional caliper, but with an angular encoder monitoring rotational displacement at the rotating joint. The instrument has three co-planar gauging surfaces, which contact the top surface of the part being gaged. Spherically tipped styli protrude an equal distance from each gauging surface. To measure the inside diameter of a tapered ring for example, the gauging surfaces of the instrument would be placed onto the top surface of the ring, and the three styli would be brought into contact with the inside surface of the cone as shown in figure 2.

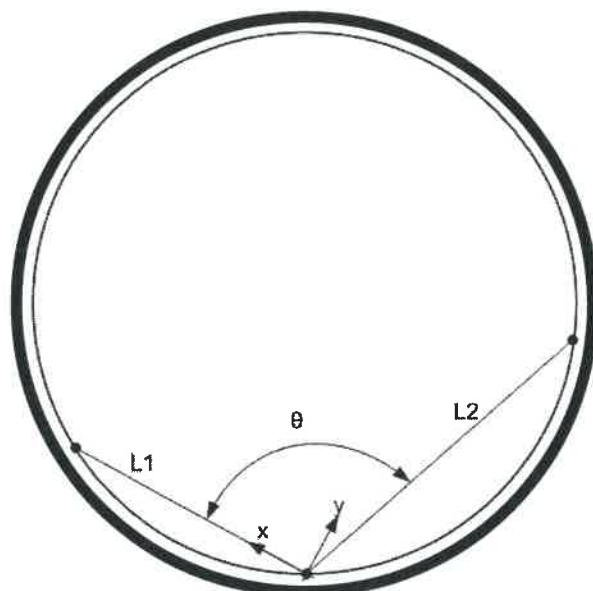


FIGURE 2. Measuring a ring using 3 point contact

By measuring the angle between the arms with the angular encoder, and assuming the arm lengths, L_1 and L_2 , and the radius of the styli are known, the ID of the tapered cone at this standoff distance from the end surface can be calculated using equation (1.1) [1].

$$D = D_{\text{styli}} + \sqrt{L_1^2 + \frac{L_1^2 \cos^2 \theta}{\sin^2 \theta} - \frac{2L_1 L_2 \cos \theta}{\sin^2 \theta} + \frac{L_2^2}{\sin^2 \theta}} \quad (1.1)$$

To measure the taper angle, the opposite side of the gauge has the styli setup with a different stand-off distance from the gauging surface than the first side (Figure 3).

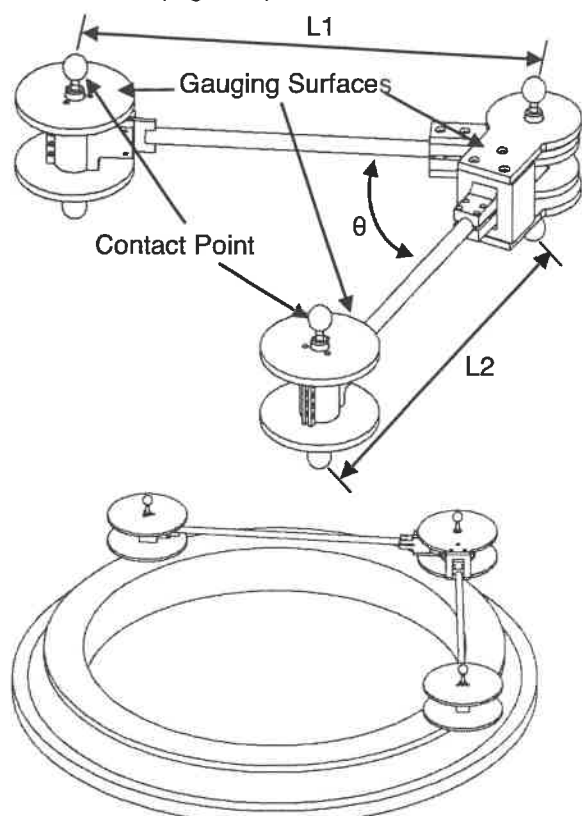


FIGURE 3. Angle encoder proportional caliper.

Knowing the difference of the stand off distances between the two sides, and the measured diameter at each height, the taper angle of the tapered ring (Figure 4) can be resolved using the following equation (1.2).

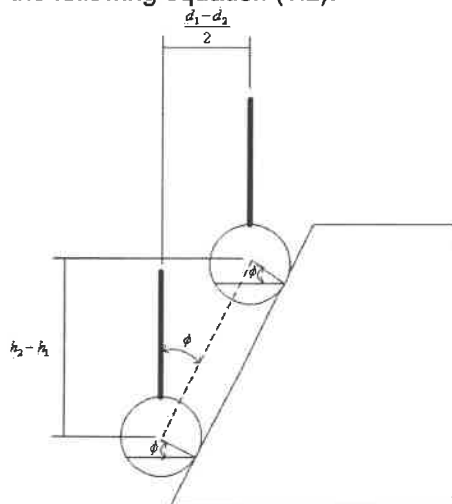


FIGURE 4. Measuring cone taper

$$\phi = \tan^{-1} \left(\frac{d_1 - d_2}{2(h_2 - h_1)} \right) \quad (1.2)$$

INITIALIZING THE INSTRUMENT

Unlike traditional bar gauging methods which transfer measurement from the master part to the part being manufactured, the angle encoder caliper doesn't require a calibrated artifact to initialize it.

The first step to setting up the instrument is to install the styli and set the proper stand off distance (Figure 5).

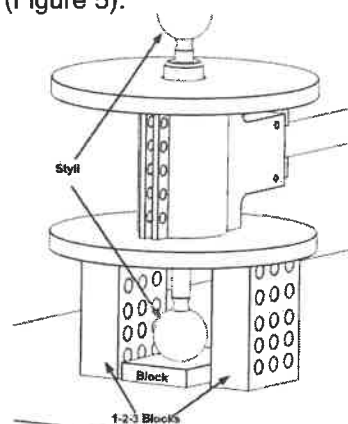


FIGURE 5: Setting the stylus standoff distance using a gage-block stack

Once all of the styli have been installed, the distance from the stylus located at the pivot joint to the styli located at the ends of the arms is measured using a coordinate measuring machine (CMM). The results of this measurement will be used as lengths L1 and L2.

Encoder systems are displacement measuring devices. In order for an encoder to make absolute measurements a datum needs to be set, and with an angular encoder, an angle of a known value needs to be introduced to it. This can be done without using a precision calibrated artifact, but using the instrument's encoder, and a stable uncalibrated artifact [2]. This method is known as self calibration [3]. To apply this method to our instrument, all that is needed is an artifact that has a means to interface with the instrument in a repeatable fashion, and in more than one position. The initialization and calibration process is outlined in Figure 6.

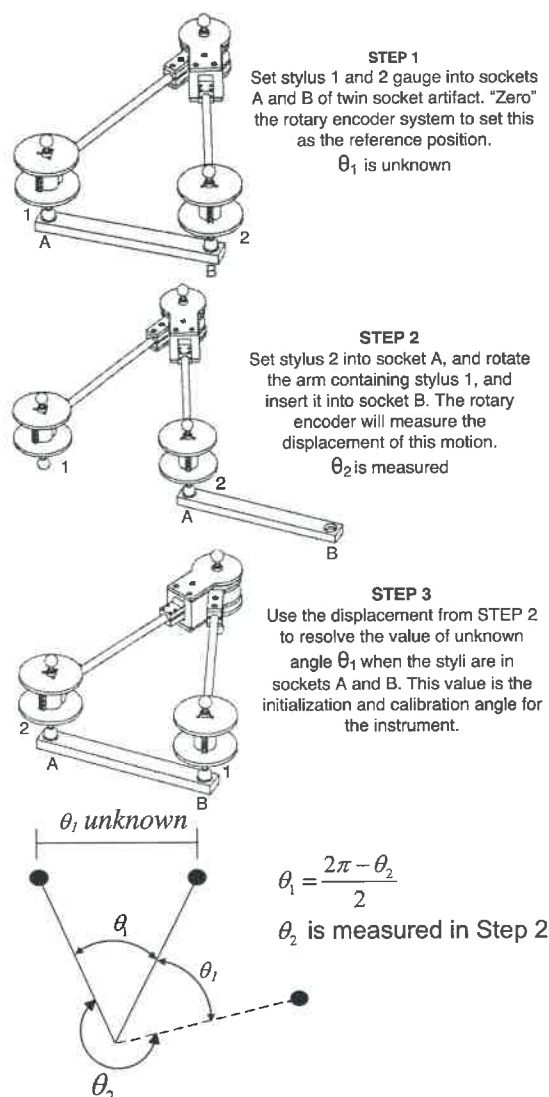


FIGURE 6: Three step initialization and calibration procedure

The greatest benefit of initializing and calibrating the instrument in this manner is that a calibrated artifact is not required. All that is required is that the distance between the two sockets on the artifact be stable during the short time required for the calibration procedure. With the instrument initialized, measurements may proceed.

UNCERTAINTY ANALYSIS

During the detailed design phase of the instrument, a measurement uncertainty analysis was performed using Type B evaluations [4]. The Type B evaluations involved in this design included items such as manufacturer's specifications for the angular encoder system,

manufacturing tolerances, uncertainty of the CMM measured length of each arm, and environmental operating conditions. There are 7 sources of measurement error that we expect to have a significant impact on the measurement; they are outlined as follows. Uncertainty contributions are computed assuming measurement of a tapered ring that is nominally 2000mm in diameter.

Qualification Error

In the qualification process of the instrument, the lengths of each of the measurement arms need to be determined. To perform this task a CMM is used to measure the distance from the stylus located at the pivot to each of the styli located at the ends of the measurement arms. We assume the CMM has a maximum measurement error of 2.0 μ m; and assuming a half width rectangular probability distribution, this gives a standard uncertainty of 1.15 μ m for the arm lengths.

Thermal Expansion of Arms

Temperature changes can cause the lengths of the arms to change. From the CMM lab to the shop floor, where the instrument will be used, we assume temperature changes of up to 2°C. The material for each arm will be constructed using carbon fiber and with an assumed nominal

coefficient of thermal expansion of $2.5 \times 10^{-6} \frac{1}{^\circ\text{C}}$ and a length of 1,000mm, the estimated standard uncertainty of the arms lengths is 2.89 μ m [5].

Radial Runout of instrument

The functionality of the instrument requires two bearings to allow one arm to rotate relative to the other. Since these bearings are not perfect there will be some radial error motion. In addition, the ball centers of each of the styli will not be perfectly concentric with the average axis of rotation. This error source will change the effective length of the moving arm. During construction the balls will be aligned such that the combined radial error motion and non-concentricity does not exceed 2.5 μ m, leading to a standard uncertainty of 1.44 μ m. (Since only one arm moves relative to the other, runout only affects one arm.)

Uncertainty of Styli ball diameter

The balls of the each of the styli are not perfect and will deviate slightly from the ideal sphere. For this design grade 3 balls will be used. The estimated standard uncertainty of the measured diameter due to spherical deviation for the stylus ball is estimated to be 44 nm.

Encoder Accuracy

A precision angular encoder is used to record the change of angle between the two arms. The data supplied by the manufacturer of the encoder ring indicates a maximum error of 1.0 arc second which translates to a standard uncertainty of 2.77×10^{-6} radians.

Initialization of angle encoder

Initialization of the instrument will be dependent on the resolution of the encoder, and the repeatability of the interface between the balls and the sockets of the artifact shown in figure 6. Each of the sockets is a kinematic socket which provides micron level repeatability. Assuming each socket has a positioning repeatability of $1 \mu\text{m}$, and with each arm nominally 1,000mm in length we expect the maximum angular error to be 2.0×10^{-6} rad, yielding a standard uncertainty of 1.15×10^{-6} rad.

Hertzian Contact Penetration

With the balls of each of the stylus contacting the surface of the part, there will be some elastic deformation at the point of contact. This deformation will cause the angle between the arms to deviate slightly, thus causing a measurement error. Assuming a contact force of 10 N and steel styli of radius 12.7 mm, we estimate an angular standard uncertainty of 8.08×10^{-7} rad.

Using this information, the expected uncertainties for measuring a 2000 mm diameter steel cone are outlined in Table 1.

The combined standard uncertainty of a diameter measurement is estimated by the following equation.

$$UD = \sqrt{\left(\frac{\partial D}{\partial L_1} UL_1\right)^2 + \left(\frac{\partial D}{\partial L_2} UL_2\right)^2 + \left(\frac{\partial D}{\partial \theta} U\theta\right)^2 + \left(\frac{\partial D}{\partial D_{stylus}} UD_{stylus}\right)^2} \quad (1.3)$$

Where $\frac{\partial D}{\partial L_1}$, $\frac{\partial D}{\partial L_2}$, $\frac{\partial D}{\partial \theta}$, and $\frac{\partial D}{\partial D_{stylus}}$ are partial

derivatives of equation 1.1 taken with respect to length of arm 1 and 2, angular measurement, and stylus ball diameter, also known as sensitivity coefficients and UL_1 , UL_2 , $U\theta$, and UD_{stylus} are combined uncertainties for length of arms 1 and 2, the angular measurement, and stylus ball diameter.

TABLE 1. Uncertainty analysis for measuring a cone.

Uncertainty Analysis	Value	Units
Uncertainty in Length of Arm 1	Std. Uncer.	
Qualification Error (measure point to point on CMM)	1.15E-03	
Thermal Expansion due to Uncer. in temp. change	2.89E-03	
Radial runout of instrument	1.44E-03	
Ball sphericity	4.40E-05	
Combined Uncertainty UL_1	3.43E-03	mm
Uncertainty in Length of Arm 2	Std. Uncer.	
Qualification Error (measure point to point on CMM)	1.15E-03	
Thermal Expansion due to Uncer. in temp. change	2.89E-03	
Radial runout of instrument	0.00E+00	
Ball sphericity	4.40E-05	
Combined Uncertainty UL_2	3.11E-03	mm
Uncertainty of Angular Measurement	Std. Uncer.	
Encoder Accuracy	2.77E-06	
Angle change due to Hertzian Contact Penetration	8.08E-07	
Initialization of angle encoder	1.15E-06	
Combined Uncertainty $U\theta$	3.11E-06	Rad
Ball diameter uncertainty	4.40E-05	mm
Expanded Uncertainty (k=2)		
Uncertainty of Diameter 1 Measurement (Upper)	1.54E-02	mm
Uncertainty of Diameter 2 Measurement (Lower)	1.44E-02	mm
Uncertainty of Taper Measurement	5.60E-05	Rad

CONCLUSION

The instrument described above appears to be capable of making measurements with low enough uncertainty to enable the proposed application. Therefore, a prototype of the instrument is currently under construction. The prototype instrument will be compared to both traditional bar gauges and to CMM measurements.

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KINEMATIC AND DYNAMIC BEHAVIOR OF A WEARING JOINT IN A CRANK-SLIDER MECHANISM

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INTRODUCTION AND SCOPE

The ability to predict the behavior of components and mechanisms is central to mechanical design, often separating success from failure. In many cases, error motions in mechanical joints can have significant effects on the kinematic and dynamic behavior of the mechanism. Recent multibody dynamics modeling research highlights the effects of joint clearance, friction, and compliance on mechanism behavior [1,2].

These effects may be exacerbated as wear occurs, affecting not only the magnitude of clearance but also the shape of the joint components. Efforts to model the coupled relationship between joint geometry, contact loads, and wear have been applied to such scenarios as scotch yoke mechanisms, cam-follower joints, and simply loaded pin-bushing revolute joints [3,4]. In some cases, no experimental validation of the modeling effort is offered, while in other cases only wear is measured while load conditions are inferred.

In this work, the joint under study is a pin-bushing revolute joint between the crank and follower links of a crank-slider mechanism. The crank-slider is chosen because it provides a planar mechanism for which kinematic and dynamic solutions are available for both idealized and realistic joint scenarios. Additionally, this mechanism has relevance to the automotive, aerospace, and heavy equipment industries. Procedures for modeling the coupled evolution of wear and dynamics for this scenario have been described by Mukras *et al* [5].

The experimental goals of this project are to:

- 1) Design and build an instrumented crank-slider test bed in order to characterize the dynamic and error motion contributions of a single wearing joint in dry sliding;

- 2) Provide a library of experimental data available for comparison with dynamic and wear models.

EXPERIMENTAL APPARATUS

The crank-slider test bed (*FIGURE 1*) is designed to isolate friction, wear, and error motions to the joint of interest in order to study the joint directly, while minimizing confounding influences from the rest of the mechanism. While the pin-bushing joint is instrumented to measure dynamic and kinematic behavior and allowed to wear, the remainder of the mechanism is intended to provide idealized mechanical behavior.

The kinematic components of the test bed can be thought of as belonging to one of two categories: those intended to supply a constant rotational crank speed and those intended to allow free motion of the slide. Motion-generating components include a DC motor, gear reducer, timing belt, flywheel (used to enable constant angular velocity in the presence of disturbances; not shown), block spindle, and aluminum crank link. The crank link has a length of 76.2 mm (3 in.) while the follower link has a length of 203.2 mm (8 in.).

The pin at the joint of interest is clamped at one end in the crank arm and is free to rotate, subject to sliding friction, within a wearing test bushing. The wear bushing is clamped in the follower link. In order to provide low friction, high joint stiffness, and minimal joint wear, the follower link and slider stage are joined by two porous carbon thrust air bushings. Similarly, the dovetail slide itself is constructed of porous carbon air bearings. Wearing test bushings are machined from virgin electrical grade polytetrafluoroethylene (PTFE).

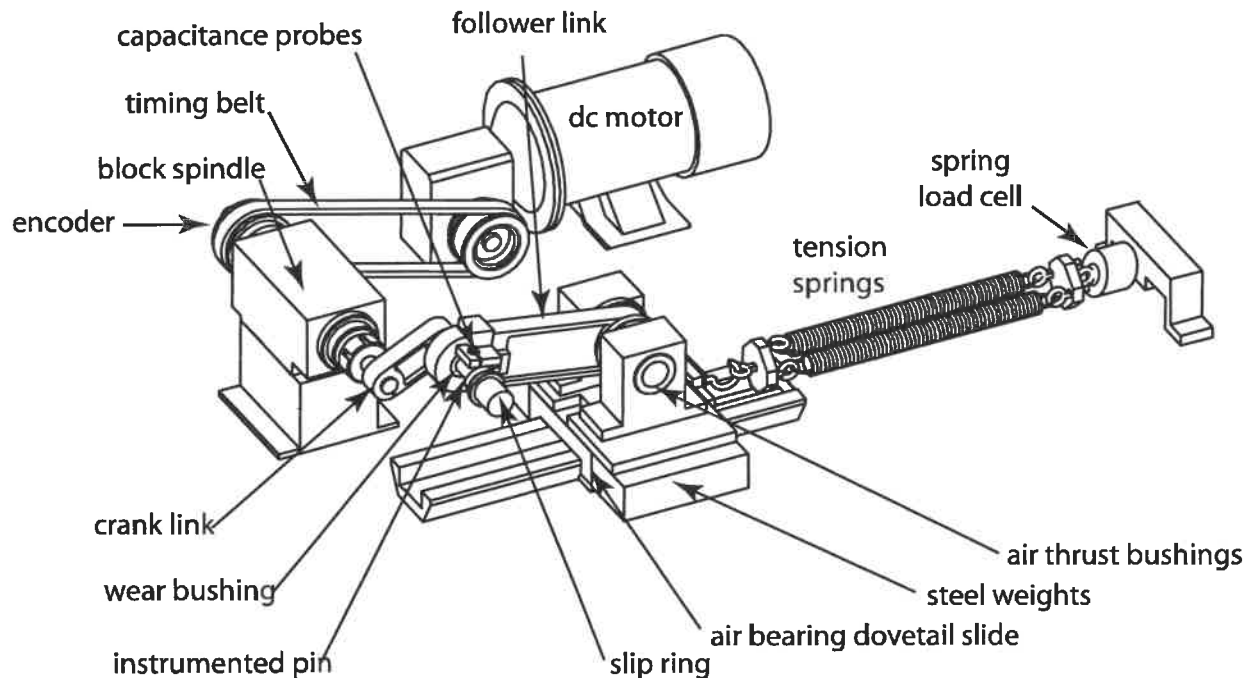


FIGURE 1. The instrumented crank-slider test bed.

The cyclic load profile at the joint under study can be manipulated in several ways. First, the mechanism inertia can be increased by adding up to 8.6 kg of steel weights to the slide stage. Second, coil tension springs can be added in parallel between the slide and the table on which the mechanism is mounted. Of course, the influence of the mechanism dynamics on the system behavior can be modified by varying the crank speed as well.

Pin forces at the joint of interest are measured using a strain gauge load cell (FIGURE 2). This load cell is composed of two full-bridge strain gauges arrays offset by 90 deg. The gauges of each bridge are arranged to measure transverse shear in a single direction while canceling the effects of axial, bending, and torsion strains. This pin is hollowed to increase strains and the gauged portion of the pin is necked to localize strains to that area. One bridge measures loads parallel to the crank radial direction, while the other measures loads parallel to the crank circumferential direction. Signals are transmitted from the strain gages to data acquisition hardware through a 10-circuit slip ring.

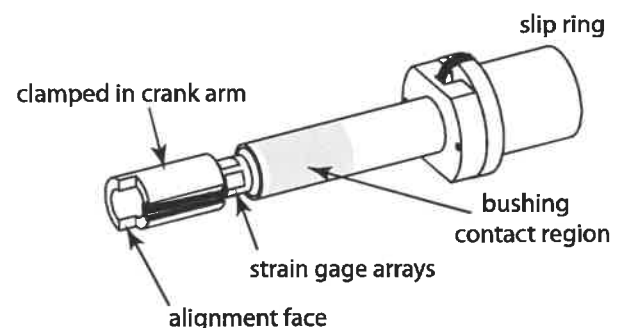


FIGURE 2. Strain gauge load cell built onto the pin at the joint of interest.

In-plane translational error motions are measured using two orthogonal capacitance probes mounted to the follower link (FIGURE 3). These probes indicate the position of the pin center with respect to the bushing center and provide a means to track the evolution of in-plane errors as bushing wear occurs.

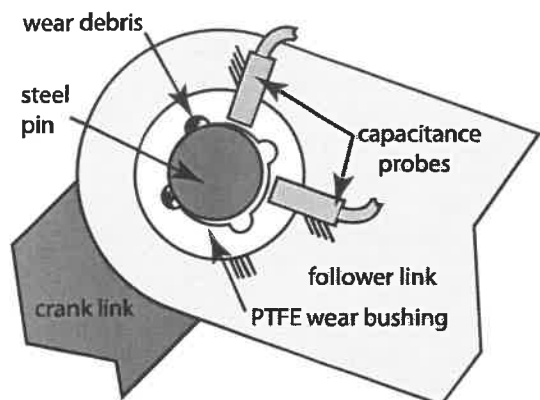


FIGURE 3. Capacitance probes used to measure the location of the pin with respect to the bushing center.

Typically, the purpose of a crank-slider mechanism is to accurately prescribe reciprocating motion of the slide due to rotation of the crank. For that reason, the position of the stage is recorded using a displacement measuring interferometer (FIGURE 4). In this setup, light emitted from the two-frequency head is split into two paths: a fixed reference path and the measurement path defined by a target retroreflector mounted on the sliding stage. The two light signals are then recombined and the interference between them is used to quantify the stage displacement.

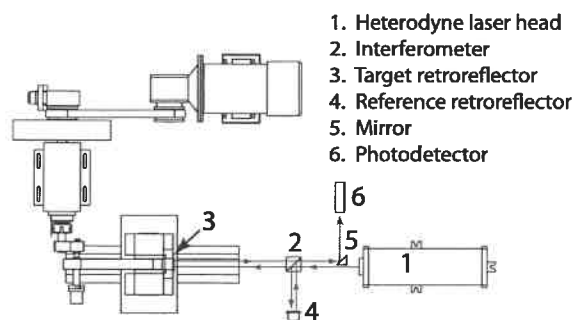


FIGURE 4. A linear displacement measuring interferometer is used to track motion of the stage.

Since the crank-slider behavior is periodic, it is often useful to plot force and displacement data with respect to the crank angle. For this reason, the angular location of the crank link is measured using a 3600 count-per-revolution hollow shaft encoder clamped to the spindle axis. The crank angle θ is measured counterclockwise from horizontal as shown in FIGURE 5.

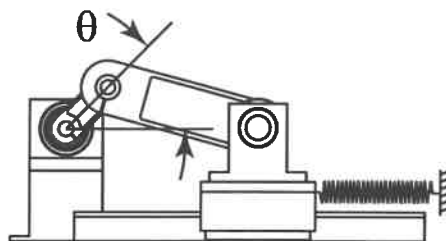


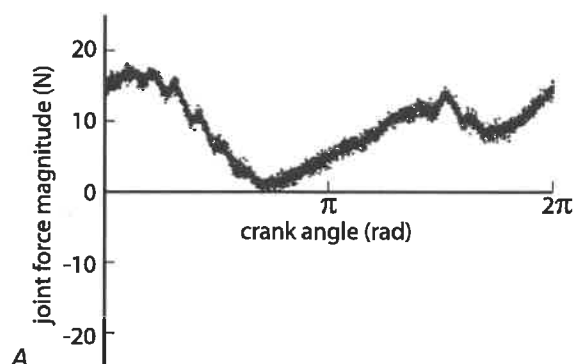
FIGURE 5. Definition of the crank angle θ .

COUPLED EVOLUTION OF WEAR AND DYNAMICS

In order to investigate the coupled nature of joint geometry and dynamics, a wear test was conducted using the conditions listed in TABLE 1. As the bushing wore over the course of the test, high frequency dynamics were seen to develop in the cyclic force profile (FIGURE 6). This content is associated with vibration of the bushing relative to the pin as the contact location shifted in direction. Similar vibration was observed in the capacitance probe and interferometer measurements (FIGURE 7).

TABLE 1. Wear test operating conditions

Initial bushing diameter (mm)	19.066
Initial bushing roundness error (μm)	22
Initial bushing mass (g)	13.1566
Crank speed (rpm)	30
Stage mass (kg)	8.5
Cycles completed	420,000



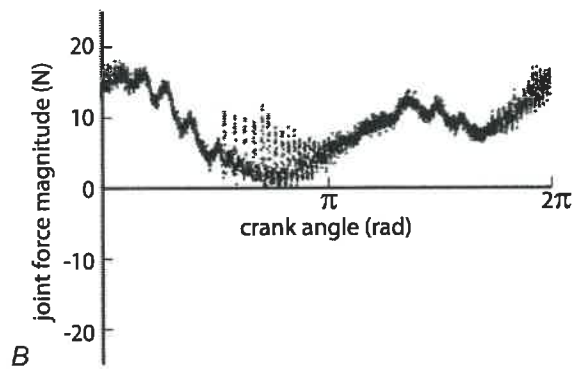


FIGURE 6. Single cycle joint force magnitude profiles at cycle 10 (A) and cycle 419,000 (B) show the evolution of high frequency vibration as the bushing wore.

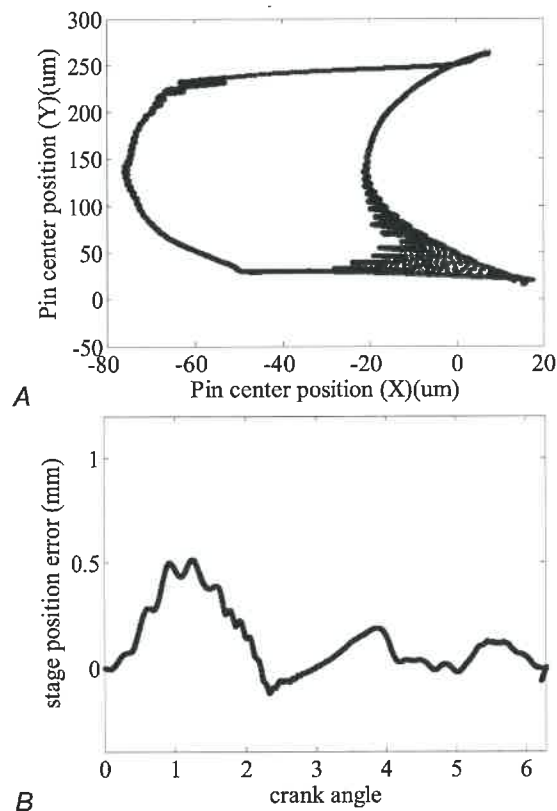


FIGURE 7. Capacitance probe measurements (A) and stage error motions measured by the interferometer (B) from cycle 419,000 also indicate the presence of high frequency vibration. The stage position error was obtained by subtracting the predicted gross motion from the measured stage position.

ACKNOWLEDGEMENTS

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LABORATORY DEMONSTRATION SYSTEM FOR PRECISION ENGINEERING EDUCATION

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INTRODUCTION

Precision engineering and metrology play a vital role in modern manufacturing, from nanotechnology to aerospace applications. Since mechanical engineers are major players in manufacturing, it is very important to introduce them to basic principles involved in precision engineering and metrology. Not only must the theory be presented, but it is crucially important that students gain hands-on experience in the laboratory in order to truly understand the fundamental concepts.

We are in the process of developing a system for instructional purposes (shown in Figure 1 below) that is capable of demonstrating some basic principles and measurement techniques involved in precision engineering and metrology.

The system consists of a monolithic single degree of freedom flexure stage that is mounted on top of a linear air bearing stage. The compound stage system serves as a combined example of the use of bearing elements that are often employed in precision engineering applications.

The system is instrumented with several kinds of displacement measurement devices, vibration, and temperature sensors.

Targets for a laser triangulation sensor and a capacitance gage are mounted on the flexure stage. Laser triangulation and capacitance probes are mounted on one side of the target. A retro-reflector for a laser interferometer is mounted on the other side of the target. The laser triangulation and capacitance sensors serve as examples of distance measuring sensors using laser triangulation and capacitance principles. The interferometer serves as an example of distance measuring technology using interferometric principles. The configuration of the triangulation sensor, capacitance sensor and the laser interferometer

is used to demonstrate Abbe's principle and what will result when there is an offset distance between the axis of measurement and the axis of translation of the stage. There are also two accelerometers, one on the flexure stage and the other on the air bearing stage. These measure relative vibration between the flexure stage and the air bearing stage. Several thermocouples are attached in different locations on the system to measure temperature. This data is used in turn to show how temperature variation affects precision measurements. The entire system is automated using a program written in LabVIEW. The system as a whole can be used to demonstrate the steps involved in the calibration of the sensor technologies. Also the system can be used for the demonstration of angular errors in a staging system.

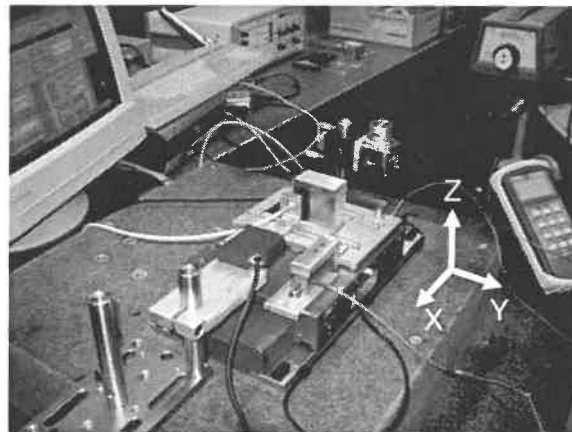


FIGURE 1. Precision engineering and metrology demonstration system. The system consists of a compound single degree of freedom flexure and linear air bearing stage. Metrology elements include a laser triangulation sensor, capacitive displacement gauge, distance measuring laser interferometer, laser vibrometer, and thermocouples for measuring ambient and system temperatures.

SYSTEM DETAILS

Stages & Motion Control

Flexure Stage

We adapted the classic single degree-of-freedom micro-positioning stage described by Scire and Teague for the flexure [1]. This was done in part for historical reasons and in part for its use of levers for magnifying the input motion provided by a piezoelectric actuator. In our experience, most mechanical engineering students have little to no exposure to flexures prior to taking our graduate level Precision Machine Design course. The use of an oft-quoted and well documented design affords students an opportunity to interact with a practical example of a monolithic flexure using circular notch hinges that they encounter in their textbook and readings for the course.

As shown in Figure 2, the flexure stage consists of circular notch hinges with an average radius of 3.36 mm and notch thickness of .64 mm and a single lever arrangement to amplify actuator input. Originally we tried to construct the cascading two-lever design described by Scire and Teague, which had an amplification factor of 28, but limitations in the output force of our piezo actuator and errors in fabrication resulted in our using only a single lever arrangement. The two lever arms measure approximately 60.0 mm and 8.3 mm, which provides an amplification of the piezo displacement of approximately 7.2:1.

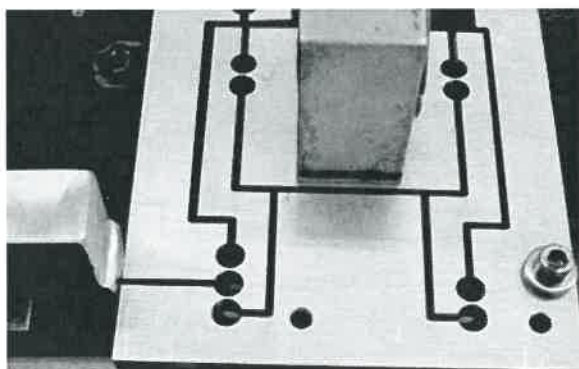


FIGURE 2. Single degree of freedom monolithic flexure stage modeled after a similar stage described by Scire and Teague, 1978. Our stage uses a single lever amplification of approximately 7.2:1. The stage is actuated by a Burleigh PZS-050 piezoelectric actuator shown in the lower left corner. The block in the center houses a retroreflector used for interferometric displacement measurement of the stage motion.

The actuator is a Burleigh model PZS-050. The piezoelectric element is specified to give an output of 50 μ m for an input of 150 VDC. We were limited to a variable DC voltage supply that was capable of supplying 0-50V, so the effective maximum travel of the tip of the actuator was approximately 17 μ m unblocked.

Air bearing Stage

The flexure is mounted on top of an Aerotech ABL 1000 linear air bearing stage, which provides both coarse and fine positioning for the entire system. Instead of the regular house air supply we used dry nitrogen at 80 psi. The encoders with the ABL 1000 allow the stage to be positioned to a resolution of 10 nm. The stages are driven using linear brushless servo motors and are controlled by Aerotech's BAI controller. The BAI controller is controlled by the system computer through LabVIEW drivers and a LabVIEW virtual instrument that we wrote.

Sensors for position measurement

Laser Triangulation sensor

The system uses a Keyence LB70 laser triangulation sensor, which has a measurement resolution of 10 microns. The measurement range for this sensor is +/- 40 mm. The manufacturer's specified linearity is 1.6% of FS (80 mm)

Capacitance sensor

The system uses an ADE 3500 capacitance displacement gauge to measure the displacement of the retroreflector as the flexure is actuated. Using a 12-bit data acquisition unit, we found the resolution of the capacitance gauge to be approximately 0.1 micron. The total range of the capacitance gauge is about 250 micron. The capacitance gauge is mounted on the air bearing stage, and its target is mounted on the back of the laser retroreflector, which is mounted on the flexure.

Laser Interferometer

The system uses an HP 5519 laser head and HP 10702A linear interferometer as the reference scale for the entire system. The retro-reflector is mounted on the center line of both the air-bearing and flexure stages. The air temperature, pressure and relative humidity are monitored by an HP 10751A air sensor, which the HP software uses for environmental correction.

Environmental Conditions

We are in the process of incorporating some monitoring of environmental conditions into the system. The effects of temperature gradients and external vibration are always of practical concern, and we would like the students to experience their effects.

Thermocouple

We used thermocouples to monitor the ambient temperature and the temperature of the stage system. We did not make use of this data in this phase of development.

Non-contact Vibrometer

We intended to also use a Polytech PDV-100 portable laser vibrometer to monitor vibration of the granite base, but we were not able to integrate this measurement into the system in the first phase of development.

SYSTEM INTEGRATION

Mechanical Hardware

The laser triangulation sensor and capacitance gauge are mounted on adjustable brackets. The capacitance gauge was purposely offset by 22 mm in the y direction (see Figure 1) from the centerline of travel of the flexure, so the effect of Abbe offset could be vividly demonstrated [2]. Figure 3 shows the difference in x-displacement measurement by the interferometer and the capacitance gage

Electrical Hardware & System Software

The air bearing stage was controlled using the Aerotech's BAI controller. The position commands were given by the program written in LabVIEW . A National Instruments USB 6008 data acquisition device was used for collecting data from the laser triangulation sensor and the capacitance gage. A LabVIEW program was written to collect this data automatically from the gages. The final goal is to integrate the motion and position feedback systems (triangulation sensor, cap gage and interferometer) using a single program to control and acquire the data automatically.

SYSTEM TESTING

It is important for students to understand that any measurement system that has been built needs to be tested for functionality and performance. Functionality tests include verifying that system components work as expected. Some examples include the working of the electrical hardware, the fit and function of the mechanical hardware, the correct

configuration of software drivers and versions, etc. Our demonstration system was tested at the individual component level, which included the stages, sensors, and data acquisition hardware to make sure each component was operational.

The main performance tests for many precision systems include tests for resolution, repeatability, accuracy, linear positioning errors, angular errors, linearity, range, etc. Due to time constraints, we chose to measure just a few combined characteristics of the stages and sensors.

The first test involved moving the air bearing stage at intervals of 5 mm over a distance of 50 mm. At every 5 mm location, the interferometer and the Keyence sensors measurements were recorded. The data from the air-bearing stage encoders were compared with the interferometer and the Keyence sensor. This experiment helps us answer two important questions:

1. Are the stage errors within the specification provided by the manufacturer? (The interferometer is assumed as the reference.)
2. Does the linearity of the Keyence sensor meet the manufacturer's specification? If not, why not? (Some reasons might include the type of the target, its location, the alignment of the sensor, etc.) Might there be other issues related to the operational characteristics of the sensor?

In our case, relating to the first question, the results in Figure 4 show that the stage errors are within the manufacturer provided specification of 0.6 micron (+/- 0.3 micron). However, the manufacturer's testing condition is unknown (location of the interferometer optics etc.)

Relating to the second question, our results showed that there is a very large error (>10%) in the measurements of the laser triangulation sensor. Other studies have shown that such errors can be due to the alignment, type of target, interaction of the sensor with the target, calibration, etc. [3] The source of the errors we found needs further investigation.

The second test involved moving the flexure stage at intervals of 5 microns over a distance of 50 microns. At every 5 micron location the interferometer and the capacitance readings were noted. The data from the interferometer was compared with the capacitance sensor data. This experiment helps us answer two important questions:

1. Are the capacitance gauge errors within the specification provided by the manufacturer? (The interferometer is assumed as the reference.)
2. What is the effect of the capacitance gauge being offset in the y direction by 22 mm from the measurement axis of the interferometer based on the Abbe principle?

The typical linearity for capacitance gauges like the one we used is about 0.25% of the range of measurement. However the data shows that these errors are of much high order, and hence the variation is due to the position of the capacitance gauge with respect to the measurement axis of the interferometer.

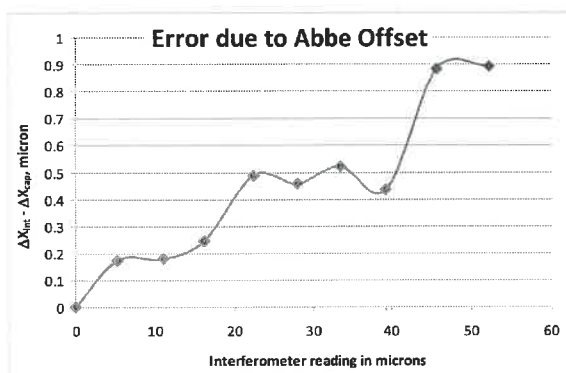


FIGURE 3. Difference in x-displacement measurement of the retroreflector as measured by the interferometer and capacitance gage. The capacitance gage axis was offset by 22 mm from the centerline of travel of the retroreflector to demonstrate the effect of Abbe offset.

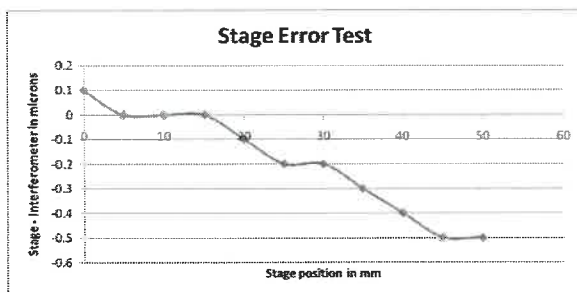


FIGURE 4. Results for investigating error in positioning the air bearing. It appears that the stage is within the manufacturer's specification of ± 0.3 micron.

CONCLUSIONS

We have created a system that can be used to demonstrate precision engineering concepts and sensor technologies and give mechanical engineering students hands-on experience with them. Specifically the system demonstrates:

- precision bearing technologies: air bearings and flexures
- the effect of Abbe offset in displacement measurement
- several displacement measurement techniques: laser interferometry, capacitive sensing, and laser triangulation

The system also allows us to demonstrate:

- concepts of range, resolution, and accuracy
- concepts of calibration
- practical aspects of data acquisition

FUTURE WORK

Future work involves making the whole system fully automated and making the fixtures so that the measurement setup can be assembled and disassembled in a very short period of time.

ACKNOWLEDGEMENTS

We would like to thank Mr. Craig Stauffer of SJSU College of Engineering Central Shop for all his help machining the flexure and fixtures in very short order, and we thank Dr. Fred Barez, MAE Department Chair for his support in acquiring the air bearing stage and its controller. We also would like to thank Dr. Bob Novotnak of Aerotech for timely help on the motion control system.

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DEVELOPMENT OF A METROLOGICAL NANOINDENTER

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INTRODUCTION

Over the past twenty-five years, and especially in the last decade, nano-scale instrumented indentation (or “nanoindentation”) has become the most widely accepted method for measurement of the elastic and plastic properties of thin films and small volumes. The widespread acceptance of nanoindentation for hardness and modulus measurement has recently been codified in recent ISO (2003) and ASTM (2007) documentary standards for instrumented indentation calibration and testing.

Nanoindentation is a primary source of experimental data that alternately inspires or supports theories of length-scale dependent phenomena, especially those that suggest that strength controlled by dislocation nucleation and motion is enhanced at the micro-scale. However, these sorts of experiments are dogged by noise, imprecision, and difficulty in instrument calibration. Marked improvements in temporal,

force and spatial resolution will have the capability to explore discrete plasticity [1,2] and nanoscale fracture phenomena [3,4] with very acute indenters [5].

The goal of the development program presented in this paper is to establish a directly SI-traceable platform for nano-scale mechanical metrology. Such a system will advance nanomechanical measurement science by reductions in force and displacement noise with accompanying fast servo loops of force and displacement using state of the art real-time and field-programmable gate array (FPGA) based control. Incorporation of the latest force and displacement sensing techniques could reveal nanomechanical phenomena (especially coupled elastic-plastic-fracture phenomena) that are not accessible to commercial nanoindentation instruments or cantilever-based nanomechanical instruments (scanning probe- or atomic force- microscopes, for example).

This paper presents preliminary studies of nanomechanical device designs for incorporation into a test-bed instrument. Each component not only provides function but contain known uncertainties or a specific methodology to ascertain uncertainty values.

TYPICAL NANOINENTER DESIGN ISSUES

Figure 1 is a simplified diagram of a nanoindenter in which the force and measurement loops follow a common path. It can be seen that there are a number of compliances around the force loop, the most obvious being frame compliance. In theory, the frame should provide high stiffness reference for instrument and be immune to any environmental change such as temperature, humidity etc. Not only typical frame deforms with applied load but because it is usually an assembly that relies on friction and applied preloads, it will, at some level of precision, contain hysteresis. Other contributions to compliance around the force

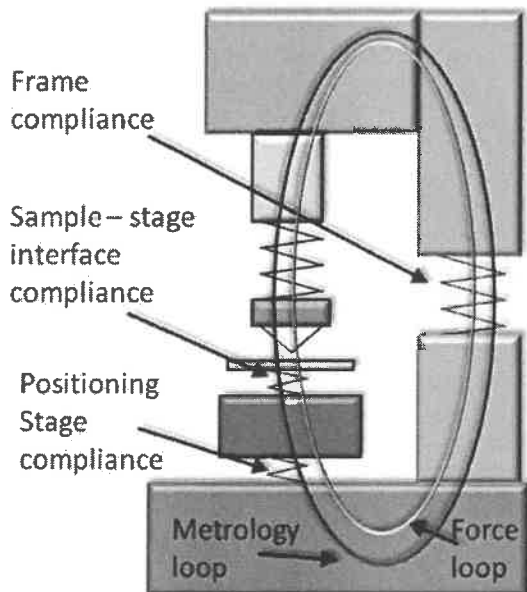


FIGURE 1: Block diagram indicating typical nanoindenter components and parasitic compliances.

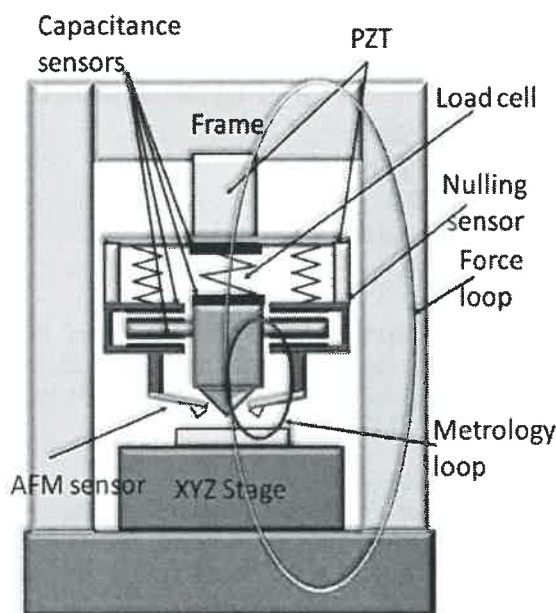


FIGURE 2: Block diagram of simplified current design showing main components and separation of force and metrology loops.

loop include the specimen, specimen to holder interface and force and displacement sensing will introduce uncertainties that may be difficult to predict.

There is also a compliance between the specimen and positioning stage table. A worst case scenario is when it is simply placed on the table. In this case it relies on three point contact formed from peaks of roughness and form of both surfaces. The contact area most probably is based on elastic averaging in an unpredictable manner which again varies with applied load and position of the indenter tip contact. A better, and commonly used practice, is to bond the specimen to a holder with glue or hot wax which forms a more linear interface characteristic. Both methods require time for the holder to reach equilibrium in the instrument.

The positioning stage and specimen holder will also add to nonlinearities of the instrument. Most positioners do not exhibit linear behavior (i.e. constant compliance) with applied load and, because they contain actuators, generate heat that will soak through the instrument causing drift during experiments.

Temperature change effects will require instrument calibration before each experiment to linearize and predict drifts which, for extremely

accurate measurements, might be neither practical nor, in some designs, possible. Wherever the metrology loop has the same path as the force loop each disturbance is directly coupled with measurement which, in most cases, cannot be extracted with predictable uncertainties.

Other than these issues that directly relate to the mechanical design of the instrument there remain a number of implementation issues that are still considered challenging for nanoindentation. Briefly, these include; prediction of areas functions for compensating indenter tip geometric effects, creating an accurate time-base for measurement processes, control stability, tip to specimen contact detection, signal to noise improvements to detect dynamic material deformation phenomena during indentation, accuracy to enable absolute parameter extraction, temperature stability and control, stiffness and force calibration, standard reference materials (particularly for softer materials) and a lack of complimentary methods to name a few.

CONCEPTUAL DESIGN

To achieve traceability each of the previously described phenomena has to be quantified. To eliminate and alleviate some of these challenges it has been necessary to design an instrument in which all significant uncertainties can be evaluated.

The principal of operation of this indenter (Figure 2) incorporates separation of force and measurement loops as far as is deemed possible. In particular, atomic force microscopy (AFM) type displacement sensing is directly referenced to the surface of the specimen. This forms a virtually rigid reference for penetration measurement of indenter tip into specimen surface while introducing force uncertainties of known value. Measurement is achieved by keeping the AFM sensor in contact with the specimen surface relatively close to indenting region but far enough that indentation induced surface deformations are not detected. Controlling its position with PZT actuator will enable maintenance of a constant force applied to the specimen surface. Ideally with sufficient sensitivity and stability of this sensing element it should be possible maintain a constant force with uncertainties being quantified from the controller error signal. By balancing between

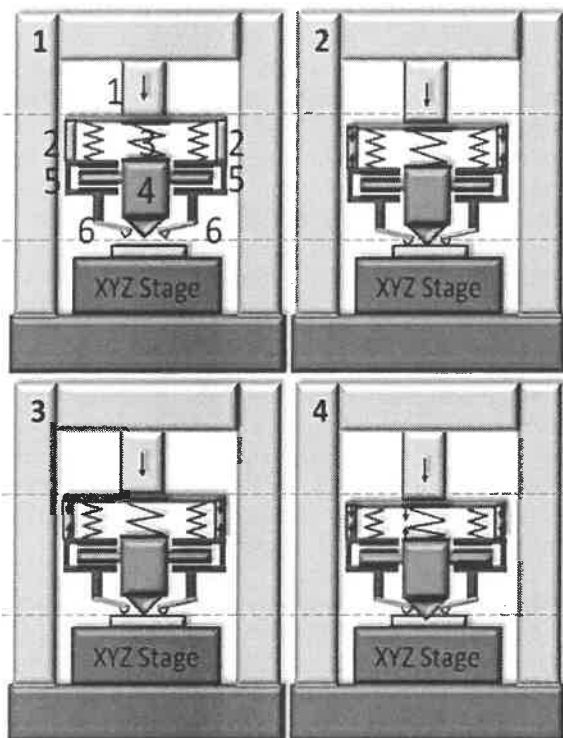


FIGURE 3: Block diagram indicating main sequence of displacement of each component during nanoindentation.

contact and adhesion, positive, negative, or even zero, forces may be used as the null value.

The mutual load between the indenter tip and surface is measured via deflection of a flexure based load cell using either a sub-nanometer resolution fiber optic interferometer [6] or capacitance gauging. The indentation depth is measured as a difference between nulling sensor and load cell position measured by double differential capacitance gauges disposed symmetrically about the load cell platform, see Figure 2.

The nanoindenter head is designed to be as symmetrical and monolithic as possible to reduce uncertainties associated with assembled structures and reduce influence of thermal distortion on instrument operation. Being in the region of the instrument where force and measurement loop coincide, the goals of the indenter tip holder are to produce stiffness values of greater than $10^7 \text{ N}\cdot\text{m}^{-1}$ with insignificant hysteresis while enabling a relatively simple indenter replacement with the

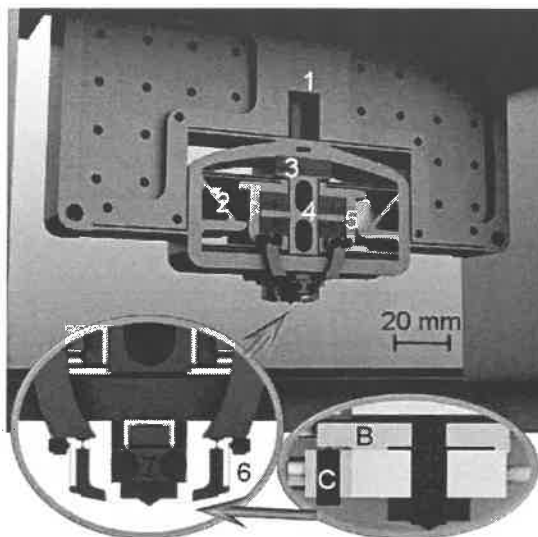


FIGURE 4: Current design with indicated basic components and cross section through indenter tip holder with preloading mechanism.

indenter tip having known orientation relative to the instrument coordinate frame.

Calibration methods will include interferometric measurement for displacement and combined interferometry and dead weight loading for force. It is envisaged that dead weight loads at nanogram levels will utilize electrolytic removal of known mass values.

PRINCIPLE OF OPERATION

The schematic representation of a typical measurement sequence is shown in Figure 3. In the first stage of motion the actuator 1 displaces the indenter frame 2 towards the specimen until AFM sensor 6 will detect contact with the surface. In the second stage while actuator 1 continues its motion, PZT 2 will contract to keep constant force applied from AFM sensors by following the specimen surface. In a third stage, the continued motion of the instrument brings indenter tip 4 into contact with specimen surface which is detected by load cell 3. Stage four is the actual indentation process where actuator 1 provides needed displacement, sensor 6 and servo loop 2 tracks the surface compensating for any stage, frame, instrument etc. parasitic deformation. The force is measured by load cell 3 and indentation displacement is measured by differential capacitance gauge 5.

The prototype design is shown in a Figure 4. The assigned numbers follow the same

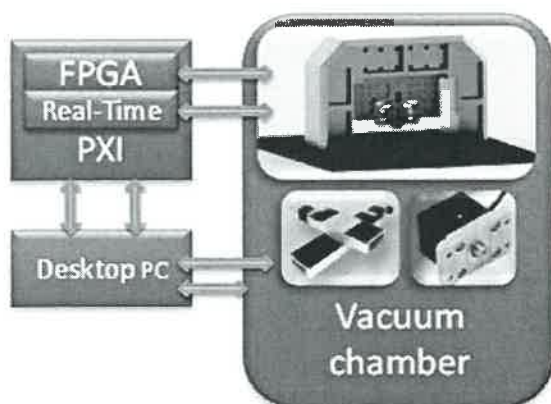


FIGURE 5: Information flow between various components of the system.

convention from Figure 3 to enable identification of the functional components of this design. The fully symmetrical design comprises of main body with pockets for associated preamplifiers and conditioning circuitry to achieve minimized signal paths. Following the numbers: 1 is main PZT, 2 is PZT for driving servo loop around nulling mechanism 5, 3 is the load cell with differential capacitance gauge and laser fiber optics laser interferometer[6] achieving picometer resolution, 6 is an extension arm of the nulling sensor that incorporates a fine soft-hard spring mechanism for adjusting the AFM tip relative to that of the indenter, 7 is the indenter tip holder with the clamp mechanism shown in bottom right corner of Figure 4. In this mechanism a typical diamond tip is attached to the threaded shaft A which is then manually screwed into nut B. The preload is then increased by the in-built leverage of the nut B with a further load imposed by the fine thread screw C. Also an adapter is available for mounting of other types of commercially available indenters.

The instrument comprises the precision indenter head and a coarse translation stage for specimen positioning. The coarse translation stage is mounted below the indenter head and labeled xyz stage, in Figure 3. This coarse stage provides a displacement range of 150 mm in X, 100 mm in Y and 5 mm in Z to enable selection of indent location and also to position the specimen vertically to bring its surface within the 10 μm range of the indenter head.

To further improve operation and stability of the instrument a vacuum chamber has been built providing seismic and acoustic isolation from the laboratory environment and further thermal

insulation to reduce temperature changes during measurement. The vacuum chamber will also provide a Faraday cage to reduce electronic noise and should further improve measurement resolution. Resolutions in the region of tens of picometers for displacement over a range of few micrometers and force measurements lower than 50 nN over a 10 mN range are expected.

Figure 5 is a greatly simplified diagram of the entire instrument. The brain of the instrument is a PXI chassis, which includes an independent microelectronic controller. Critical measurement timing and data collection are controlled by a FPGA card operating at a 40 MHz clock speed enabling event timing at these speeds and hundreds of kHz for PID control. Less critical calculations will be managed by PXI real-time with 18-bits A/D such as control of nanopositioning motor for a coarse Z displacement. All other operations such as control of vacuum system, precision XY stage and cameras are carried out using a host computer.

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A MICROACTUATOR BASED ON AN INCHWORM WITH FRICTION CONTROL DEVICE

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INTRODUCTION

The objective of this work is to develop a microactuator which is a small inchworm-like robot with a simple, modular and flexible design. This microactuator moves accurately and precisely in several centimeters to a meter square area. The microactuator can be used in precise measurement, positioning, modification, and so on. For example, the microactuator with a small proximity displacement sensor approaches a target in order to measure the length of an object. The microactuator equipped with a small tool which can make small holes or can finish surface moves closer to the target surface. A cooperative work by the microactuators with different sensors and actuators is suitable for constructing a small-scale manufacturing system [1].

INCHWORM

The inchworm-like microactuator is a minimalist robot which is designed to imitate the movements of the inchworm caterpillar [2]. The function of the inchworm-like microactuator is realized by two main components. One is the attachment mechanism and the other is the transferring mechanism. The attachment mechanism allows the microactuator adhere to the surface and provide the anchoring force needed to support the displacement. The transferring mechanism thrusts the attachment mechanism sequentially. In the conventional inchworm-like microactuator, an electromagnetic force, electrostatic force, or mechanical force comprises the attachment mechanism. A piezoelectric actuator is usually used for the transferring mechanism. The piezoelectric actuator is a transducer which converts electrical energy into a mechanical displacement using a piezoelectric effect. The piezoelectric actuator has the advantage of a high actuating precision and a fast reaction.

The inchworm-type microactuators we previously developed consisted of the electromagnets for the attachment mechanism and the piezoelectric actuators for the transferring mechanism [3]. The electromagnet which generates the electromagnetic force is an inductive load, and the piezoelectric actuator which realizes the minute displacement is a capacitive load. Since the microactuator used the electromagnets and the piezoelectric actuators, we have to prepare a high current energy source for the electromagnets and a high voltage energy source for the piezoelectric actuators. From the view point of energy source, it is desirable to use one energy source.

The linear motion realized by the microactuator was disturbed by the friction between the surface and the microactuator. The friction caused the microactuator to meander on an operation surface. The experimental results showed poor repeatability. These are caused by the effect of the uncertain friction. It is really important to eliminate the friction. Levitation is one solution which makes friction-free conditions. Electromagnetic levitation, electrostatic levitation, and acoustic levitation are methods by which an object is suspended with no support mechanism. Since piezoelectric actuators are used to transfer the microactuator, the levitation using ultrasonic vibration caused by the piezoelectric actuators is used.

DESIGN

Figure 1 shows the concept of an inchworm-like microactuator with friction control devices. The microactuator consists of friction control devices and push-pull devices connected alternately. The friction control device is used as the attachment mechanism, which activated by vibrating a stacked-type piezoelectric actuator

with high frequency. The vibration reduces the friction force or the friction-free conditions are realized. The push-pull devices are also stacked-type piezoelectric actuators which deform in the horizontal direction. In Figure 1, the extension of PP2 which is expressed with the suffix "e" moves FC3, while FC3 is activated. Then PP1 pushes and PP2 pulls FC2. The push-pull motion sequentially transfers the friction control devices.

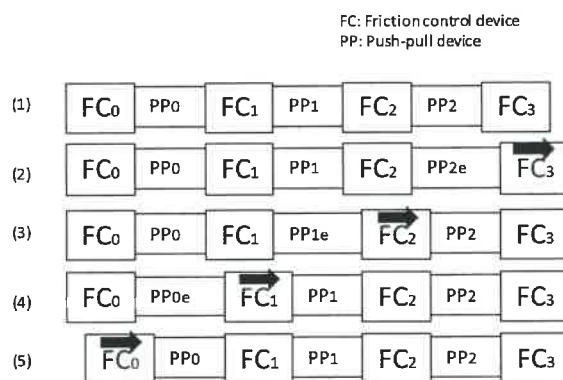


FIGURE 1. The concept of the microactuator with friction control devices. Activated friction control (FC) devices are moved by push-pull (PP) devices sequentially. Lower case letter "e" denotes the PP device with extension.

Figure 2 shows the schematic diagram of the control of the components of the microactuator. The friction control device is activated sequentially and the push-pull devices move the friction control device between them. The attachment force of the activated friction control device is reduced.

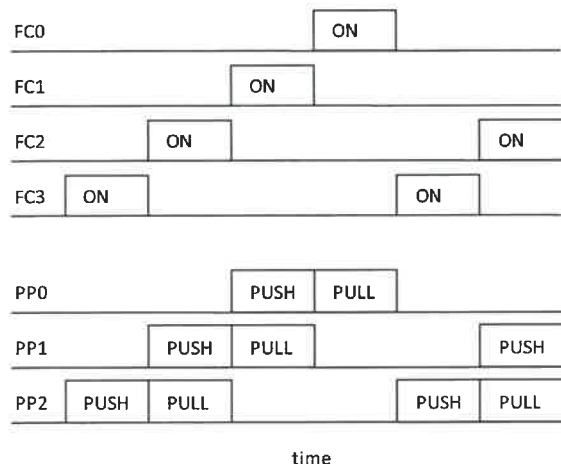


FIGURE 2. The schematic diagram of the control waveform.

PRINCIPLE OF FRICTION CONTROL DEVICE

Figure 3 shows the principle of a friction control device. The structure is simple. Three weights are connected by a vertical piezoelectric actuator (V-PZT) and a horizontal piezoelectric actuator (H-PZT). Since the friction is proportional to the weight, the friction force FC is smaller than the friction force FL which is obtained while V-PZT does not vibrate as shown in Figure 3(a). The vertical vibration of V-PZT can reduce the friction force, and the friction force FS is smaller than the constant friction force FC in Figure 3(b). In Figure 3(a), Weight 1 moves to the left when the H-PZT extends. In Figure 3(b), Weight 2 and 3 move to the right when the H-PZT extends.

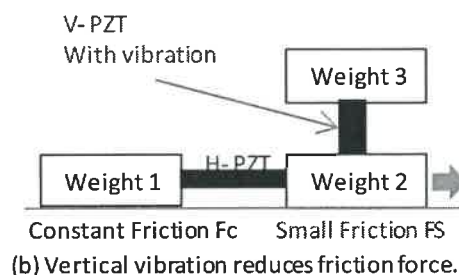
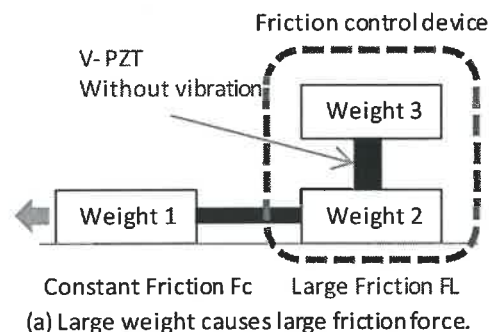


FIGURE 3. The friction control device. Constant friction force, small friction and large friction determine the moving direction.

EXPERIMENT

Figure 4 shows the preliminary experimental setup. Weights, about 0.03 kg, are connected with piezoelectric actuators. A vertical piezoelectric actuator connects two weights. It deforms 60 nm/V. A horizontal piezoelectric actuator connects two weights with an elastic hinge which isolates the vertical vibration caused by the vertical piezoelectric actuator.

The horizontal piezoelectric actuator deforms 110 nm/V.

Optical displacement sensors are used to measure the displacement of the components. The horizontal displacement X , the displacement of weight in the z direction, and the displacement of the vertical piezoelectric actuator in the z direction are measured by the displacement sensors.

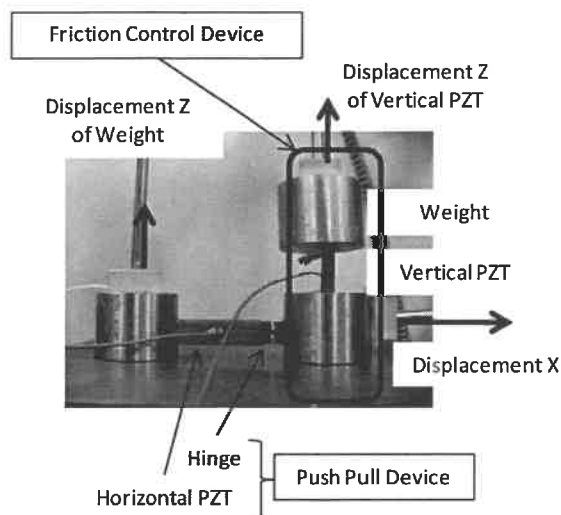


FIGURE 4. A fundamental microactuator with friction control device moved by push-pull device.

The push-pull piezoelectric actuator deforms with a sinusoidal signal instead of the triangle waveforms illustrated in Figure 5. The friction control device is controlled by a sinusoidal waveform with amplitude modulation. While the amplitude of the vertical vibration is large, the signal is expressed "ON". The friction force F_S is small or ideally becomes zero. Small amplitude of the vertical vibration large friction force (FL). The friction control device is vibrated with about 14 kHz, which is determined experimentally. The control frequency of the horizontal piezoelectric actuator is 5 Hz. The frequency of the amplitude modulation is also 5 Hz. The phase between the "ON" period of the friction control device and the deformation of the push-pull device determines the moving direction as shown in Figure 5.

Figure 6 shows experimental results in the X -direction. Control waveforms described in Figure 5 are applied to the friction control device (the vertical piezoelectric actuator, V-PZT) and the push-pull device (the horizontal piezoelectric actuator). The Z -displacement of the V-PZT is

synchronized with the control signal. While the amplitude of the V-PZT is large, the friction force is reduced and constant friction force F_c supports the microactuator. The friction control device therefore moves in the X direction. The Z displacement of weight is reduced by the use of the elastic hinge.

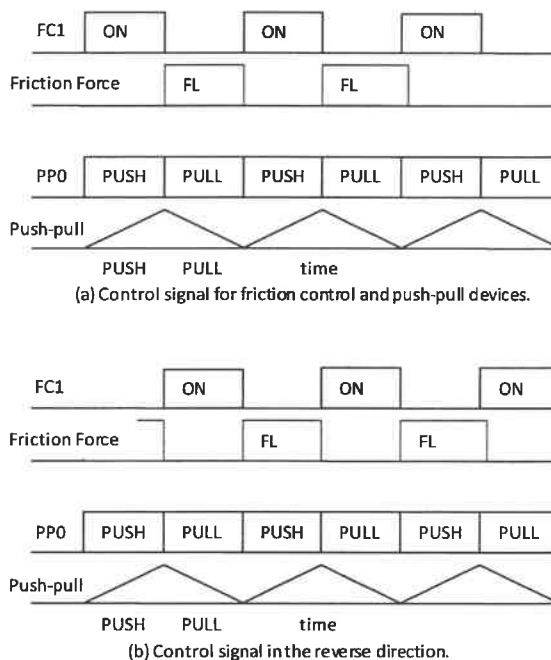


FIGURE 5. Control waveforms to the microactuator.

Changing the phase of the H-PZT and the V-PZT may reverse the moving direction. The experimental result however does now show the reverse linear displacement in Figure 7. The microactuator vibrates at a fixed position and sinusoidal waveform can be seen. The asymmetrical structure of the microactuator may cause this result.

SUMMARY

Preliminary experimental results obtained by the minimalist configuration of the inchworm-type microactuator are described. The microactuator described in this paper has two-types of devices, moving device and friction control device. Both devices are comprised of piezoelectric actuators which vibrate at the appropriate frequencies. The moving device is operated at low frequency and the friction control device is vibrated at high frequency with amplitude modulation.

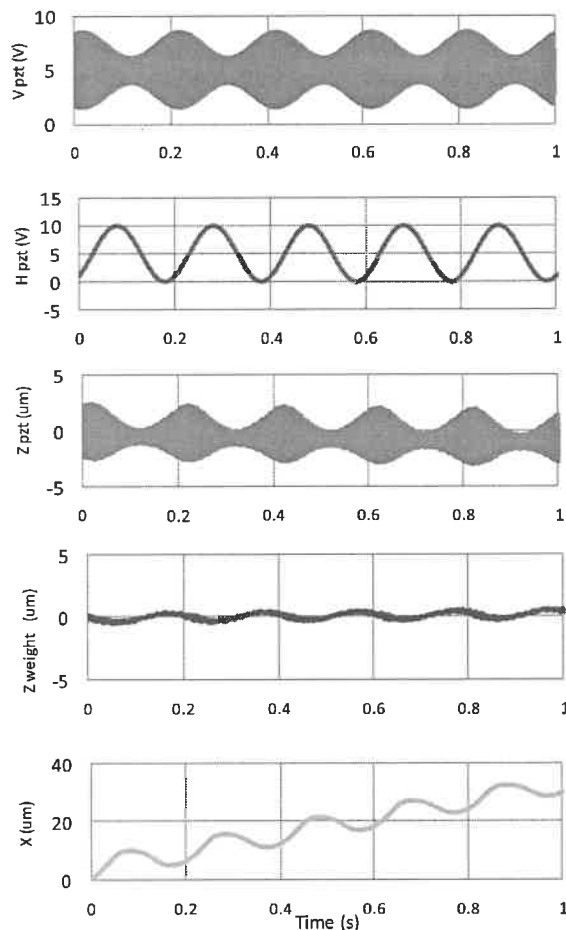


FIGURE 6. Experimental results (I). The microactuator can move by the use of sinusoidal waveform for the horizontal piezoelectric actuators and amplitude modulated sinusoidal waveform for the vertical piezoelectric actuator.

The possibility of the linear displacement was indicated. Since all the device is actuated by the piezoelectric actuator which is a capacitive-type load, we prepare one type of energy source although conventional inchworm-type actuator requires two types of energy source.

FUTURE WOKRS

This paper described the preliminary and fundamental experimental results. In another application, the friction control devices and displacement devices are connected in a straight line, and the linear displacement is demonstrated. Two-dimensional alignment of these devices is going to show a linear and rotational motion on a surface.

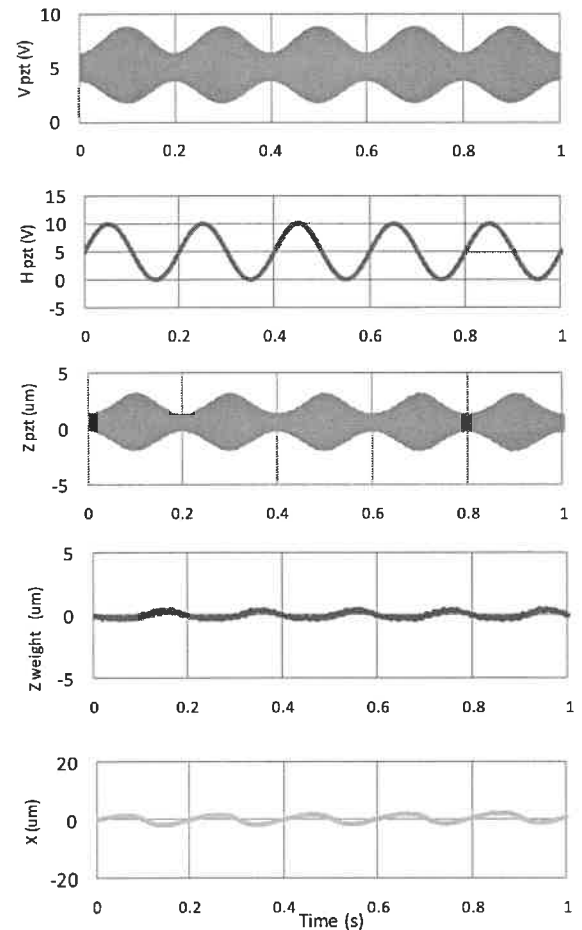


FIGURE 7. Experimental results (II). The microactuator cannot move. The microactuator vibrates at a fixed position. The phase between the V-PZT and H-PZT differs from Figure 6.

ACKNOWLEDGEMENTS

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ERROR BUDGET OF ON-MACHINE MEASUREMENT SYSTEM FOR MICRO/NANO MILLING PROCESS

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INSTRUCTION

On-machine measurement system (OMMS) is essential to guarantee the machining accuracy. The traditional doubt to the feasibility of an OMMS is that the measured data will inevitably include the motion error of the machine tool since the same machine is used as the datum of measurement. Although various OMMS have been studied in literatures, most of them are implemented without considering this doubt theoretically. Therefore, this doubt prevents the OMMS to be accepted and widely used in the industrial fields. The objective of this paper is to present the methodology to evaluate the feasibility of the OMMS on a micro/nano machining center.

MOTIVATION TO IMPLEMENT OMMS

During the last decades, a dramatic increase of number of high accuracy micro components for electro-mechanical instruments, aerospace equipments and medical devices can be observed. Micro/nano mechanical machining with conventionally shaped tools has gained increasing importance since it is suitable for fabrication of 3D micro component made from engineering materials.

During micro/nano milling process, the requirement of the geometric accuracy is at sub-micron level. It is especially difficult to control the geometric accuracy in the sub-micron level, due to many different sources of error. Even if the geometrical error of a machine tool is compensated, the inaccuracy machining of parts often occurs due to other factors related to machining process such as the bending of cutting tools, inaccuracy of tool geometry and setting, thermal effects and the inappropriate cutting process design, etc.[1-2]. A state of the art machining center, Sodick's AZ150 high speed and ultra precision milling center, has a programmable resolution of 0.01 micron.. However, the dimensional deviation of machined part sometimes exceeds 1 micron. In order to verify and control the machining quality of parts,

the measuring system is imperative for micro/nano machining. Currently, the measurement and geometry verification of micro parts can only be done on off-line measurement instruments such as SEM (Scanning Electron Microscope), WLI (White Lighting Interferometer), etc. These off-line measurement systems can achieve high measurement resolution of micro/nano scale. However, they cannot prevent the inaccuracy completely during machining except to verify it after machining. In addition, no efforts can be done to correct the dimensions to the desirable ones even if the workpiece is found to be undercut. The reason is that once the fixture is released, it is difficult to set the part back for compensational machining operations.

In order to solve these problems, the best way is to measure the geometrical errors by using OMMS after a semi-finishing process. Then the measurement results are analyzed and used to modify the tool path of a finishing process. However, the on-machining measuring system based on this methodology is not available for micro/nano machining. Although the in-process measurement of workpiece geometry has a rather long history development, promising methods to meet the requirement from industry have been rarely seen. Lack of OMMS has become the bottleneck to increase the efficiency and reduce the cost of micro/nano machining. Therefore, it is necessary to develop the OMMS and research on the theory of machining error compensation during micro/nano machining process.

FEASIBILITY OF OMMS

A non-contact displacement sensor with nano meter resolution will be installed into the machine tool to establish the OMMS on a Sodick AZ150 micro/nano machining center (Figure 1). The AZ150 is a three axis, linear-drive milling machine with movement resolution of 10nm. There is contact probe on the machine tool for workpiece setup. The same position to install the

contact probe can be used to install the displacement sensor as shown in Figure 2.



FIGURE 1. AZ150 Machine Tool.

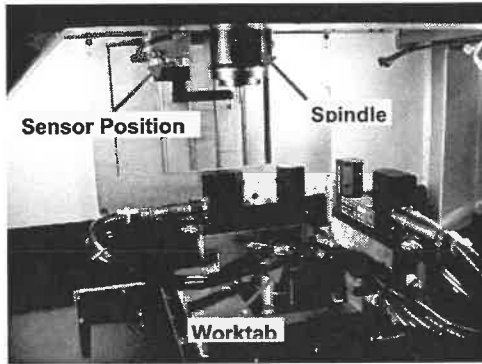


FIGURE 2. Position to Install Sensor.

Traditional doubt to the OMMS is that the part is measured on the machine, using the same machine motion by which it was made. The axis of machine tool is used as datum of OMMS. Then, the measured data will be inevitably includes positioning error from the inaccurate axis motion of the machine tool. In order to theoretically evaluate the feasibility of OMMS, the following approach is used. First, the error model of OMMS is constructed; then, based on this model, the performance of OMMS is simulated for a given measurement task to evaluate the feasibility of OMMS.

ERROR MODEL OF OMMS

The kinematic modeling technique [3] can be used to establish the error modeling of OMMS under the rigid body assumption. A machine tool is viewed as a series of kinematic links between bodies. Each pair of links can be described by the coordinate transformation. The kinematic model of a machine tool can be established by the sequential matrix multiplications of all the homogeneous transformation matrices (HTM). The volumetric error model of a machine tool involved two kinematic links, e.g., the tool branch and the part branch. For the OMMS, the

sensor branch is involved. Figure 3 show the kinematic links of AZ150 micro/nano machining center.

The X axis assembly has only one degree of freedom, so the ideal HTM between CS₀ and CS₅ is

$${}^0T_5 = \begin{bmatrix} 1 & 0 & 0 & x \\ 0 & 1 & 0 & Y_5 \\ 0 & 0 & 1 & Z_5 \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad (1)$$

Y_5 and Z_5 are constant offset. x is the servo-controlled motion along x axis.

The actual HTM is

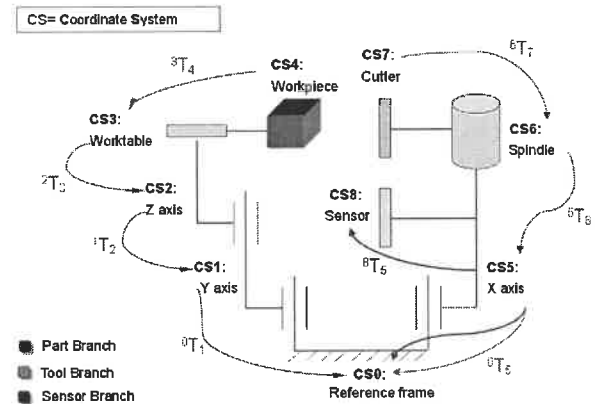


FIGURE 3. Kinematic Links of AZ150.

$${}^0T_{5error} = \begin{bmatrix} 1 & -ECX & EBX & EXX + x \\ ECX & 1 & -EAX & EYX + \alpha_{xy}X + Y_5 \\ -EBX & EAX & 1 & EZX + \alpha_{xz}X + Z_5 \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad (2)$$

Where, EXX is the positional deviation of the X axis. EAX , EBX , and ECX are the roll, pitch, and yaw error of the X axis respectively. EYX and EZX are the linear deviations of X axis in Y and Z direction respectively. α_{xy} is the orthogonality error between X axis and Y axis. α_{xz} is the orthogonality error between X axis and Z axis.

The sensor is rigidly attached to the X axis, so the ideal HTM between CS₅ and CS₈ is

$${}^5T_8 = \begin{bmatrix} 1 & 0 & 0 & X_8 \\ 0 & 1 & 0 & Y_8 \\ 0 & 0 & 1 & Z_8 \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad (3)$$

However, the actual HTM is

$${}^5T_{\text{error}} = \begin{bmatrix} 1 & 0 & \varepsilon_y(\text{se}) & X_8 \\ 0 & 1 & -\varepsilon_x(\text{se}) & Y_8 \\ -\varepsilon_y(\text{se}) & \varepsilon_x(\text{se}) & 1 & Z_8 \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad (4)$$

where, $\varepsilon_x(\text{se})$ and $\varepsilon_y(\text{se})$ are the angular errors of the sensor around X axis and Y axis respectively.

An important measurement tasks in a micro/nano machining process is to verify the height or width of a critical feature such a thin wall or step on a micro part. The measurement of height for a thin wall structure is used to evaluate the feasibility of OMMS on AZ150.

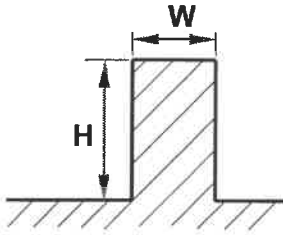


FIGURE 4. Measurement Task.

Only 1D servo-controlled motion is involved to measure the height. Suppose it is X axis. The measurement process is illustrated in Figure 5. The sensor moves along X axis. Only use the measured distance by the sensor in the beginning and end position. In Position1, the measured distance is H_1 . Then move the sensor to Position2 along X axis. The measured distance is H_2 . The coordinate of point a in sensor coordinate system is $(0, 0, H_1, 1)$. The coordinate of point b in sensor coordinate system is $(0, 0, H_2, 1)$. Their ideal coordinates with respect to reference frame are

$$A = {}^0T_5 \cdot {}^5T_8 \cdot a = \begin{pmatrix} x_a + X_8 \\ Y_5 + Y_8 \\ H_1 + Z_5 + Z_8 \\ 1 \end{pmatrix} \quad (5)$$

$$B = {}^0T_5 \cdot {}^5T_8 \cdot b = \begin{pmatrix} x_b + X_8 \\ Y_5 + Y_8 \\ H_2 + Z_5 + Z_8 \\ 1 \end{pmatrix} \quad (6)$$

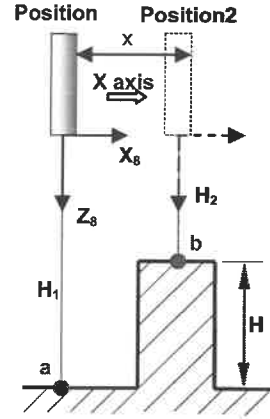


FIGURE 5. Measurement Process.

The ideal vertical distance between a and b is:

$$H = A(3) - B(3) = H_1 - H_2 \quad (7)$$

While in the actual case, the sensor itself (distance measuring uncertainty, orientation error) and the machine axis will introduce errors. The actual coordinate of point a in sensor coordinate system is $a_{\text{error}}(0, 0, H_1 + \delta_1, 1)$. δ_1 is the measurement error of sensor at position1. The actual coordinate of point b in sensor coordinate system is $b_{\text{error}}(0, 0, H_2 + \delta_2, 1)$. δ_2 is the measurement error of sensor at position2. Their actual coordinates in reference frame is

$$A_{\text{error}} = {}^0T_{\text{sensor}} \cdot {}^5T_{8\text{error}} \cdot a_{\text{error}} \quad (8)$$

$$B_{\text{error}} = {}^0T_{\text{sensor}} \cdot {}^5T_{8\text{error}} \cdot b_{\text{error}} \quad (9)$$

The actual vertical distance of A and B is

$$H_{\text{actual}} = A_{\text{error}}(3) - B_{\text{error}}(3) \quad (10)$$

The absolute measuring error is

$$E = H - H_{\text{actual}} \quad (11)$$

SIMULATION

Equation (11) can be used to simulate the measurement error for this measurement task. The range of each error component is given in Table 1, which has uniform distribution. The position of x axis (X_a and X_b), the measured data from the sensor (H_1 and H_2) as well as the constants (Y_5, Z_5) is also given.

A Monte-Carlo simulation is used. According to the range of each error component, a random number is generated to represent a possible value of this error component. Then equation (11) is used to simulate the possible measurement error of the OMMS for a given

task. 10000 times of simulations are conducted to simulate the measurement error. As shown in Figure 6, the standard deviation is 294nm, which can be used to represent the measurement uncertainty.

TABLE 1. Error components and constants involved in measurement.

CS5 (X axis)		
x_a	2500 μ m	motion along x axis at position A
x_b	3500 μ m	motion along x axis at position B
Y5	75mm	Constant offset
Z5	150mm	Constant offset
EXX(x_a)	[-50,50]	positional deviation of X axis in X direction
EXX(x_b)	nm	
EYX(x_a)	[-50,50]	the linear deviation of X axis in Y direction
EYX(x_b)	nm	
EZX(x_a)	[-50,50]	the linear deviation of X axis in Z direction
EZX(x_b)	nm	
EAX(x_a)	[-1,1]	the roll error of X axis (around X axis)
EAX(x_b)	arc sec	
EBX(x_a)	[-1,1]	the pitch error of X axis (around Y axis)
EBX(x_b)	arc sec	
ECX(x_a)	[-1,1]	the yaw error of X axis (around Z axis)
ECX(x_b)	arc sec	
α_{xy}	1 arc sec	the orthogonality error between X and Y axis
α_{xz}	1 arc sec	the orthogonality error between X and Z axis
CS8 (Sensor)		
X8	50mm	Constant offset
Y5	50mm	Constant offset
Z8	-50mm	Constant offset
ε_x (se)	1 arc sec	the angular error of tool around X axis and Y axis
ε_y (se)		
Measurement point in coordinate frame CS8 is a [0, 0, $H_1 + \delta_1$, 1], b [0, 0, $H_2 + \delta_2$, 1]		
H1	300 μ m	The ideal distance at the first position
H2	150 μ m	The ideal distance at the second position
δ_1	[-50,50]	Length error due to tool setting
δ_2	nm	

CONCLUSION

The typical measurement task is analyzed. The nominal height is 150 micron. The movement distance along X axis during measurement is

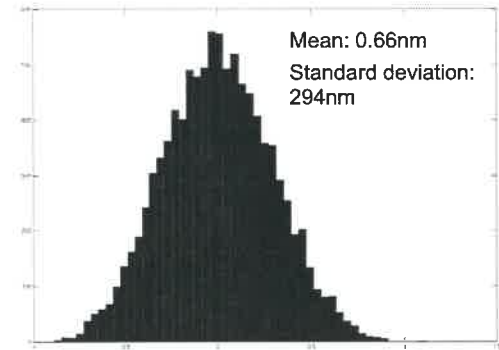


FIGURE 6. Simulation Result.

1.5mm. For a displacement sensor with 50nm uncertainty, the overall measurement uncertainty of OMMS on a micro/nano machining center AZ150 is around 300nm. The measurement uncertainty is acceptable to guarantee sub-micron machining tolerance. The sensor with such accuracy is available on the market. Therefore, to establish the OMMS on AZ150 to ensure the sub-micron machining accuracy is feasible.

The conclusions support the fact that on the micro/nano machining center, measurement condition is better than machining condition. First, the motion resolution is very high (AZ150 is 10nm), which is the same and even better than most of offline measurement system. Second, main part of machining error comes from machining process. For example, reference [4] points out that tool wear and tool deformation due to the process forces themselves is the main source of error in micro-milling processes.

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IMPROVING THE ACCURACY OF A 6-DOF DISPLACEMENT/ORIENTATION SENSOR FOR HEAVY TRUCK K&C TESTING

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INTRODUCTION

Kinematics and Compliance (K&C) testing is vital to the understanding of the elasto-kinematic behavior of tire and suspension systems for passenger cars and light and heavy trucks. Michelin currently operates the only facility in North America capable of performing K&C tests on heavy trucks. In K&C testing, horizontal and vertical loads are applied to the chassis and/or tires, resulting in 6 degree-of-freedom (DOF) motion of the wheel relative to the chassis. This motion is composed of 3 components of translation along the Cartesian vehicle axes, and 3 components of rotation about those axes. Michelin measures this generalized motion using ten custom designed, un-actuated, serial (R-R-P-R-R-R), 6-DOF displacement transducers, hereinafter referred to as the Universal Plane Transducer Assembly (UPTA). (Figure 1).

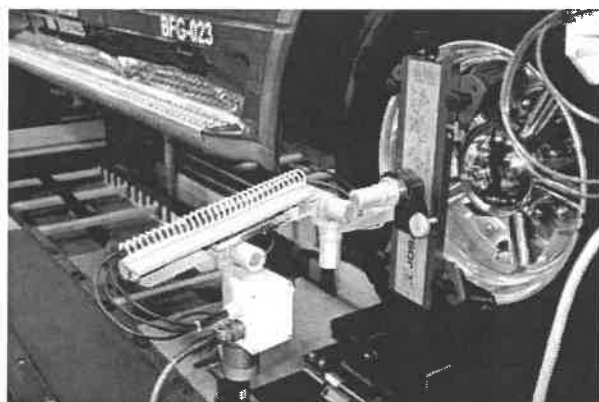


FIGURE 1. UPTA fixed to a truck wheel.

The UPTA axis encoders measure changes in the rotary or linear position of its joints, and this data is transformed into Cartesian translations and rotations of the moving end using a kinematic model of the mechanism. Due to inevitable manufacturing and assembly tolerances, the as-built device does not conform perfectly to the design drawings and so a simple kinematic model (SKM) based on the nominal dimensions of the device results in measurement errors. To rectify this, an

Advanced Kinematic Model (AKM) containing 42 unknown model parameters was developed. This paper describes how the AKM was developed, as well as two calibration methodologies for identification of the model parameters. The target measurement accuracy for the UPTA is ± 0.1 mm in position and $\pm 0.05^\circ$ in orientation.

KINEMATIC MODELING

Simple Kinematic Model (SKM)

The SKM treats each joint of the UPTA as a perfect rotary or linear joint whose axis is exactly perpendicular to, and intersect, the axes of the adjacent joints in the chain. The axis encoders of the UPTA do not provide absolute position and do not have index marks to establish a "home" position. Therefore, prior to use, the UPTA must be inserted into an initialization stand of known dimensions to place all joints in a known and repeatable position. Subsequent joint motions are then added to these known initial positions. (Figure 2)

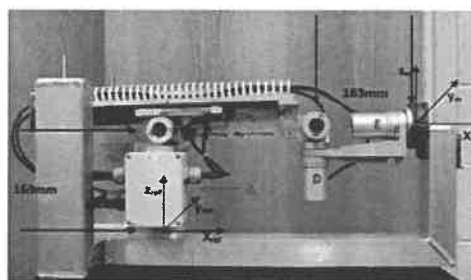


FIGURE 2. UPTA in initialization stand

Homogeneous Transformation Matrices (HTMs) are used to establish the SKM that relates the base coordinate system to the moving end.

$$SKM = {}^{ref}T_0 * {}^0T_1 * \dots * {}^6T_m$$

Where rotations take the form:

$${}^5T_6 = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & \cos(\theta_6) & -\sin(\theta_6) & 0 \\ 0 & \sin(\theta_6) & \cos(\theta_6) & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

and θ_6 is the measured encoder value of joint 6. Translations take the form:

$${}^3T_4 = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ 0 & 0 & 1 & z_3 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

and z_3 is the measured value of encoder 3.

Advanced Kinematic Model (AKM)

The AKM does not make the same geometric assumptions as the SKM. In the AKM, it is assumed that adjacent joint axes may deviate by small amounts from being perpendicular, and may not intersect. Therefore, an error transformation matrix, E_i , is used to describe these small errors.

$$E_i = \begin{bmatrix} 1 & -\gamma_0 & \beta_0 & \Delta x_0 \\ \gamma_0 & 1 & -\alpha_0 & \Delta y_0 \\ -\beta_0 & \alpha_0 & 1 & \Delta z_0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

where α , β , γ , Δx , Δy , Δz represent small angular and translational deviations of the coordinate axes and origins from the nominal. The corrected HTM for each axis is the product of the nominal and error HTM.

$${}^0T_{1,corr} = {}^0T_1 * E_1 = \begin{bmatrix} c\theta_1 & -s\theta_1 & c\theta_1\beta_1 + s\theta_1\alpha_1 & c\theta_1\Delta x_1 - s\theta_1\Delta y_1 \\ s\theta_1 & c\theta_1 & s\theta_1\beta_1 - c\theta_1\alpha_1 & s\theta_1\Delta x_1 + c\theta_1\Delta y_1 \\ -\beta_1 & \alpha_1 & 1 & \Delta z_1 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

The AKM contains 42 unknown parameters, and is given by:

$$AKM = {}^{ref}T_{1,corr} = {}^{ref}T_{0,corr} * {}^0T_{1,corr} * \dots * {}^6T_{m,corr}$$

PARAMETER IDENTIFICATION

The 42 unknown parameters of the AKM are identified using a CMM to measure the pose (position and orientation) of the device in a number of configurations. The base of the UPTA is fixtured to the CMM table and because the UPTA is a passive device, an X-Y-Z- θ stage is used to move it to different poses and hold it there while the CMM measures the pose. To

facilitate pose measurement, several tooling balls are attached to the base and end mounting couplings. (Figure 3)



FIGURE 3. UPTA being measured by CMM

Using the stage, the UPTA is manipulated into 27 different poses that span the normal working space of the device. At each pose, the CMM measurements are used to compute the HTM relating the base and measurement end of the UPTA. These are compared to the HTM predicted by the AKM, and result in a residual, Φ , at each measured pose. Since the translational and rotational errors have different units, a scaling factor, X , is used to weight their relative contribution.

$$\phi = \Delta x^2 + \Delta y^2 + \Delta z^2 + \chi(\Delta \alpha^2 + \Delta \beta^2 + \Delta \gamma^2)$$

The parameter identification algorithm then uses a non-linear least squares optimization scheme to find the set of 42 parameters that minimizes the sum of squares of the residuals over the measured poses. Experiments were performed to investigate how the choice of the scaling factor affected the outcome. It was found that use of a large scaling factor improved the orientation measurement accuracy of the device without significantly degrading the position measurement accuracy. (Figure 4)

The calibration procedure was repeated for all ten of the UPTA devices. The results are summarized in Figures 5 and 6, showing an average 88.5% improvement in position measurement accuracy, and 75.3% improvement in orientation measurement accuracy.

RAPID PARAMETER IDENTIFICATION

The procedure for parameter identification described above provided excellent results. However, it requires access to a CMM, and substantial measurement time to collect the

data. To improve this aspect of UPTA calibration, an alternate method for data collection, called the Stick Calibration method, was investigated. It is a variation of the Single Point Articulation Test (SPAT) described in ANSI/ASME B.89.4.22M standard for performance evaluation of articulated arm coordinate measuring machines[3].

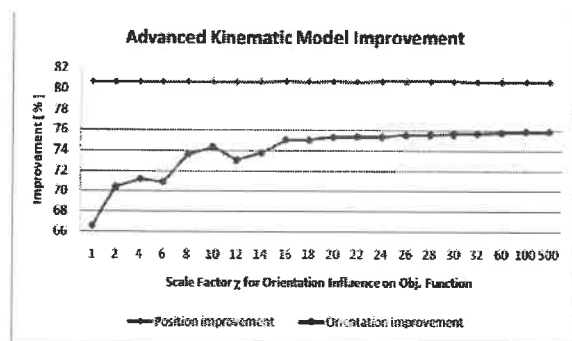


FIGURE 4. Effect of scaling factor on AKM performance

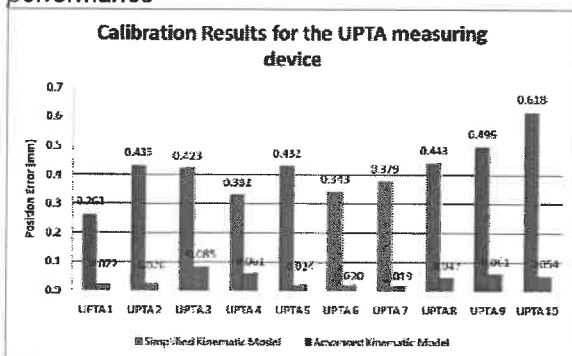


FIGURE 5. Position error comparison of SKM and AKM.

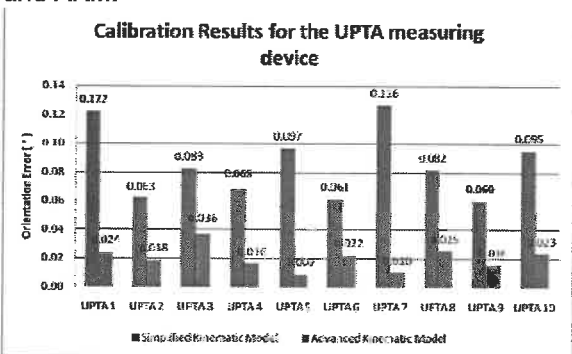


FIGURE 6. Orientation error comparison of SKM and AKM.

In that test, the probe ball of the AACMM is engaged with a kinematic mount rigidly attached to the base of the machine and the links and joints of the machine are exercised. Since the probe position did not move, the probe coordinates predicted by the device should be

constant. The range of actual values obtained is a measure of the machines accuracy. In contrast to an AACMM which typically only outputs probe position but not orientation, the UPTA must measure both position and orientation of the object it is attached to.

The stick calibration uses a rigid rod with a ball on one end engaged in a kinematic socket mounted to the base similar to the SPAT test, and the other end rigidly connected to the measurement end of the UPTA. (Figure 7) The stick is measured in a traditional CMM to provide the coordinates of the ball relative to a coordinate system defined by the end coupling, and the measurement stand is also measured to find the coordinates of the ball socket relative to the UPTA base frame.

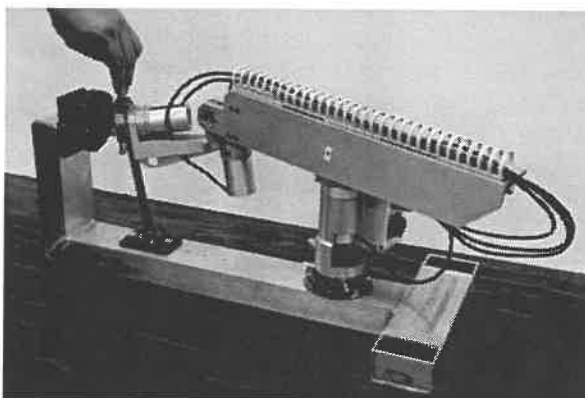


FIGURE 7. Stick calibration method

Data is collected by manually articulating the UPTA while the ball is engaged in the socket and continuously collecting joint encoder data. The ball coordinates relative to the UPTA base reference frame are then:

$${}^{ref}\vec{P}_{ball} = AKM * {}^m\vec{P}_{ball}$$

The difference between the predicted and actual ball coordinates for each pose is a residual, and the sum of squares of residuals is processed in the same manner as the previous CMM data to obtain a set of 42 kinematic parameters that minimize the residual error.

The stick calibration method does not explicitly address the orientation measurement capability of the UPTA in its parameter identification, and there was some concern whether this lack might lead to a degradation of orientation measurement accuracy. However, in practice the stick calibration method produced results

equal to or better than the CMM method and with much less effort required for data collection.

COUPLING REPEATABILITY

The UPTA devices were originally designed identical couplings on the base and measurement ends to allow them to be rapidly and easily inserted into the initialization stand and then into the vehicle measurement rig. The original coupling design consisted of a cylindrical boss on one side that inserted into a close tolerance hole on the other. A shoulder on the cylinder defined the axial position, and an alignment pin inserted into a mating hole defined the angular position. From a Precision Engineering perspective, these couplings are severely overconstrained and cannot be expected to provide good repeatability.

The influence of the base and end-effector couplings was tested by mounting the UPTA into the initialization stand and zeroing the encoders. The UPTA was then removed from the stand, moved around arbitrarily and remounted in the stage multiple times. Each time the encoder values were recorded and used with the kinematic model to predict equivalent position and orientation errors. For these trials, indicated position varied by up to 0.48 mm and indicated orientation by up to 0.23 degrees, which is well outside the target values of ± 0.1 mm for position and ± 0.05 degrees for orientation. To remedy this situation, a new Quasi-Kinematic Coupling (QKC) for the UPTAs was designed and built. (Figure 8)



FIGURE 8. Quasi-kinematic coupling

The QKC consists of two components, one of which is permanently attached to the base or measurement end of the UPTA, and the other attached to ground or the vehicle. The portion attached to the UPTA has 3 cylinders with spheres embedded in the ends. The spheres provide targets for probing by a CMM and define the base and end effector coordinate systems. The mating component is a fairly thin cylinder with V-grooves cut into it at 30° increments to enable coupling in multiple orientations. The V-grooves provide nominal line contact with the

cylinders resulting in quasi-kinematic constraint. The coupling repeatability tests were repeated using the QKC. The results showed that the QKC provided positioning repeatability less than 6 micrometers and orientation repeatability less than 0.005 degrees, thus providing nearly a 98% repeatability improvement.

CONCLUSION

The measurement accuracy of a custom designed 6-DOF serial measuring arm was investigated, and several steps were taken to improve its performance. These included redesigning the couplings used to fixture the instrument during measurement to operate in a kinematically deterministic manner to improve their repeatability, resulting in a 98% reduction in coupling repeatability.

An advanced kinematic model of the device containing 42 unknown and constant kinematic parameters was developed, and two different data collection methods were used to provide data for parameter identification, which was accomplished using a non-linear least squares criterion. The advanced kinematic model resulted in an 88.5% improvement in position measurement accuracy and 75.3% improvement in orientation measurement accuracy compared to a simple kinematic model based on nominal design dimensions.

ACKNOWLEDGEMENTS

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DIAMETER AND FORM MEASUREMENT OF A MICRO-HOLE IN A FUEL INJECTOR NOZZLE WITH THE NIST FIBER PROBE

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INTRODUCTION

We discuss the measurement of diameter and form of micro-holes with reverse tapers in a fuel injector nozzle using an ultra-low force contact fiber probe on a Coordinate Measuring Machine (CMM). The fiber probe was designed and built at the National Institute of Standards and Technology (NIST) and has demonstrated extremely low uncertainties (under 100 nm, $k = 2$) on high-quality artifacts. We present diameter and form data, and detail an uncertainty budget for the diameter. The dominant terms in the uncertainty budget are related to the poor surface finish and form of the part. Our uncertainty budget indicates that the diameter of a suitably clean micro-hole in a fuel nozzle can be determined to an uncertainty of 196 nm ($k = 2$) with our probing technique. This uncertainty is primarily related to imperfections in the hole geometry. The measurement principle of the fiber probe and validation results is described in [1]. Fig. 1 shows a schematic of the fiber inside the hole and Fig. 2 shows a photograph of the same. Prior work in this area has been reported [2-6], but their uncertainties appear larger than that from our method.

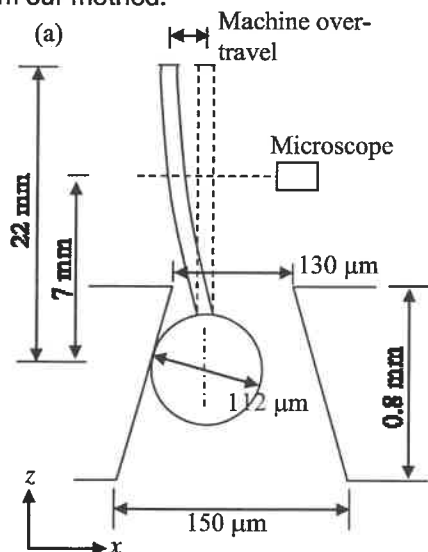


FIGURE 1 Schematic of the fiber probe in the hole

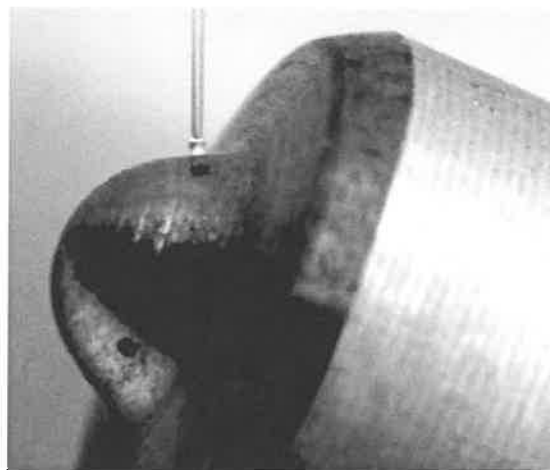


FIGURE 2 Photo of probe entering hole

DIAMETER AND FORM RESULTS

We measured circular traces at 16 different depths inside a hole, and sampled 48 equally spaced points (as 3 sets of 16 points to reduce drift effects) around the circumference at any given depth. We repeated the measurements 6 times and report the average of the 6 measurements as the diameter, see Fig. 3. A least-squares best-fit reference circle is used to compute diameter and radial form.

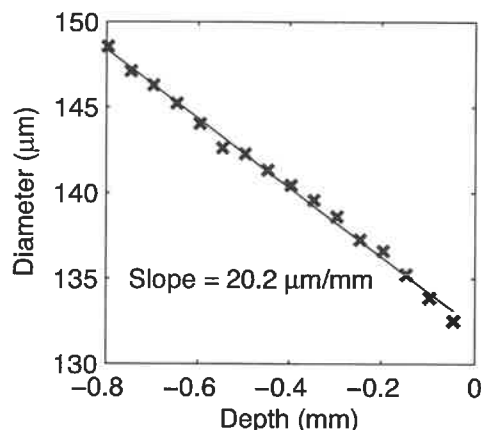


FIGURE 3 Diameter as a function of depth

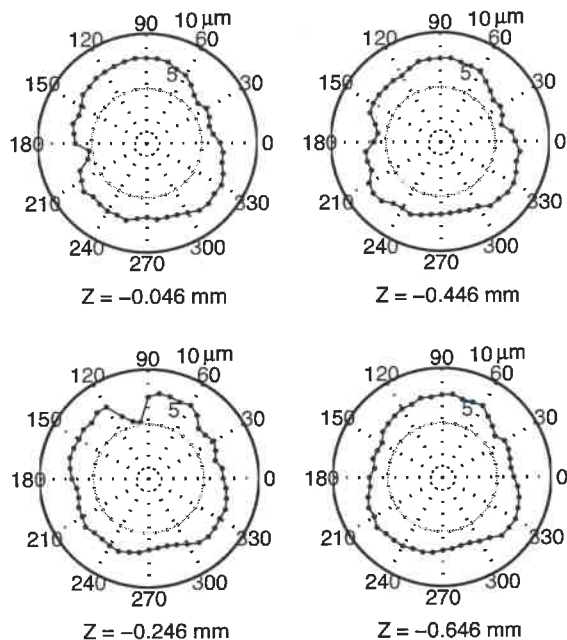


FIGURE 4 Radial form charts at some of the depths

The radial deviation chart is shown in Fig. 4 for some of the z positions. The radial out-of-roundness is between $1\text{ }\mu\text{m}$ and $3\text{ }\mu\text{m}$ for the traces. At several depths, the data shows some features that may possibly be particles of dirt or other machining residue, for example, between 90° and 120° at $z = -0.246\text{ mm}$.

UNCERTAINTY BUDGET FOR DIAMETER

The uncertainty budget for diameter measurements [7] is shown in Table 1, followed by a description of the components. The expanded ($k = 2$) uncertainty in diameter is 196 nm . This does not include the effect of dirt or machining debris that may be present. We discuss the effect of dirt on uncertainty in diameter at the end of the next section.

DESCRIPTION OF COMPONENTS

Machine Positioning and Imaging

There is an uncertainty in determining any coordinate in space due to machine positioning uncertainty and imaging uncertainty of the probing system. This uncertainty, 35 nm , impacts every measured coordinate and therefore the diameter of the measured artifact. For a 48 point measurement, the 35 nm uncertainty in a single point measurement results in a 4 nm uncertainty in the determination of diameter. This diameter uncertainty is present for both the master ball and test-hole

measurements, and therefore the combined standard uncertainty is 6 nm on diameter.

TABLE 1 Uncertainty budget for diameter

Source	Standard uncertainty (nm)
Machine and probe related	
1. Machine positioning and imaging	6
2. Determining scale factor	14
3. Axis out-of-squareness	5
Part related	
4. Tilt of the hole	1
5. Taper of the hole and error in locating top surface	24
6. Part form and finite sampling	70
7. Surface finish and mechanical filtering	50
Measurement process related	
8. Determining equatorial plane of master ball	1
9. Master ball diameter uncertainty	5
10. Reproducibility of check sphere measurements	30
11. Repeatability of hole diameter measurements	20
Combined standard uncertainty ($k = 1$)	98

Determining Scale Factor

There is an uncertainty in determining the scale factor for the probe (the nominal scale factor is $0.1\text{ }\mu\text{m}$ per pixel with $0.0001\text{ }\mu\text{m}$ per pixel uncertainty). For typical operating parameters of the probe, we determine the error in diameter due to this source to be about 10 nm . Because we measure both the master ball and the hole, the combined standard uncertainty in diameter is 14 nm .

Axis Out-of-squareness

The axes of the probing system may be non-orthogonal and misaligned with the machine's axes, and this leads to an error in the measured diameter. The error is larger when the probe size is comparable to the hole size. The nominal misalignment angles of the probing system during the hole measurements were less than -2° (with x axis) and 3.0° (with y axis). Simulations indicate that the error in the diameter of a $130\text{ }\mu\text{m}$ hole measured using a probe of diameter $112\text{ }\mu\text{m}$ is of the order of 10 nm . We correct the measured data (software

compensation) for the non-orthogonality and misalignment, and the residual standard uncertainty in diameter is determined to be less than 5 nm.

Tilt in the Hole Axis

The hole is aligned so that there is less than 2 μm tilt over 1 mm. This translates to less than 1 nm error in diameter of the hole at any given depth.

Taper of the Hole and Error in Locating Top Surface

The diameter of the hole changes at the rate of 20.2 $\mu\text{m}/\text{mm}$. Therefore, a 1 μm error in locating the reference point translates to an error of 20 nm in diameter at any given depth. Assuming a $\pm 2 \mu\text{m}$ uncertainty in locating the z reference and a uniform distribution of errors within this range, the standard uncertainty in diameter is 24 nm. It should be noted that in the normal operation of our probe, we can measure z coordinate to well under 100 nm. But for the measurements reported here, we employed a coarse positioning method of determining z coordinate, and combined with the surface geometry and roughness of the nozzle, we can only detect z coordinate to about $\pm 2 \mu\text{m}$.

Part Form and Finite Sampling

A finite sampling strategy in combination with unknown part form leads to an uncertainty in the measured diameter. We have measured 48 sampling points for each circle. Shown in Fig. 5 is the Fourier transform of the part profile. It is apparent from the figure that the dominant harmonics are of low order and are captured by the 48-point sampling strategy.

We have measured the 48 sampling points as three sets of 16 points, and individually determined the least-squares best-fit diameter for each set of 16 points. The pooled standard deviation for an individual 16-point measurement is 120 nm. If this is modeled as a random error, then the uncertainty of a 48-point measurement (three 16-point measurements) will be

$(120 \text{ nm})/\sqrt{3} = 70 \text{ nm}$. This value would somewhat overestimate the true uncertainty if there are non-random form errors at the 16th harmonic that are reduced by more than a factor of $\sqrt{3}$ at the 48th harmonic. As described below, there is some evidence from Fourier analysis that this may be the case, but we will

nevertheless make the conservative assumption that the errors are completely random. In fact, this assumption is consistent with what we expect if we model higher-order errors as due to random surface variations at each point, where the standard deviation of the distribution is determined from analysis of straightness measurements at high spatial frequencies (straightness measurements are not described in this paper). Thus we may take 70 nm as an estimate of the error arising from imperfect form in combination with finite sampling.

Fig. 5 shows results of a Fourier analysis of the form error. We see that the amplitude of higher order harmonics, say over 15 undulations per revolution, is less than 50 nm. We may reasonably extrapolate to conclude that higher order harmonics, especially the 48th harmonic (which cannot be detected by a 48-point sampling strategy), to be of small magnitude, under 50 nm amplitude.

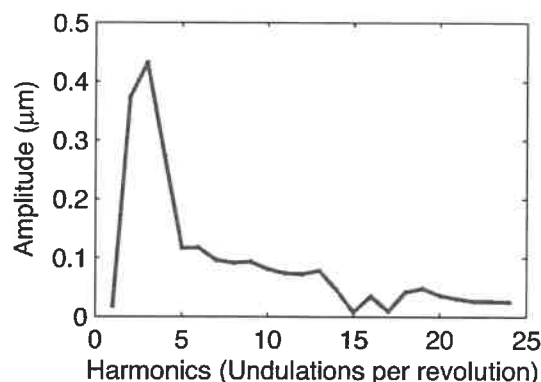


FIGURE 5 Fourier analysis of the average of the sixteen 48-point profiles

Surface Finish and Mechanical Filtering

We have not performed measurements with probes of different sizes and therefore do not have experimental data on the change in diameter due to probe size. Prior studies on other artifacts have indicated that this effect may be of the order of 50 nm. We therefore assign this value as the standard uncertainty in diameter due to mechanical filtering.

Determining Equatorial Plane of Master Ball

Probe ball diameter calibration requires accurately determining the equatorial plane of the nominal 3 mm diameter calibration ball. We can determine the equatorial plane to $\pm 1 \mu\text{m}$, and therefore the uncertainty in diameter is about 1 nm. The check sphere is larger and

therefore the determination of its equatorial plane is not as critical.

Determining Master Ball Diameter

The uncertainty in master ball diameter is 5 nm. Master ball diameter measurements are performed at NIST using the Strang viewer instrument.

Reproducibility of Check Sphere Data

The long term reproducibility of check sphere (7 mm diameter) measurements has a standard uncertainty of 30 nm. The measurements that make up long term reproducibility sample different table positions, different probe ball sizes etc. It should be noted that part of this term, approximately 10 nm, duplicates other terms already described elsewhere in this budget, but is of sufficiently small magnitude that "double-counting" of the error has negligible effect on the overall result.

Hole Diameter Repeatability

The pooled standard deviation on diameter from the 6 runs at the different depths is approximately 48 nm. The standard deviation of the mean of the 6 runs is therefore 20 nm.

Effect of Dirt, Machining Debris, and Other Protrusions

In some instances, for example in Fig. 4 at $z = -0.246$ mm, we have noticed that the part profile shows sharp inward protrusions. It is not clear if such protrusions are real surface features (part form error) or dirt/machining debris. If such features are indeed machining debris/dirt, then the computed diameter is in error and the uncertainty in diameter will be larger than that reported earlier. Assuming there is a $2\text{ }\mu\text{m}$ diameter particle spanning two sampling points (somewhat similar to that in Fig. 4, $z = -0.246$ mm), then this particle will cause a diameter error of about $0.167\text{ }\mu\text{m}$. Summing this term in quadrature with the previously reported standard uncertainty of 98 nm, we obtain a combined standard uncertainty of 194 nm, or an expanded uncertainty ($k = 2$) of 388 nm.

Dirt can easily be the largest contributor to uncertainty in diameter. Dirt in the hole will not only increase measurement uncertainty but will also give rise to a bias in the final measurement result. Rigorous application of the GUM would require correcting for this bias, but it is difficult to justify such a correction as we cannot say with certainty that the sharp features that we observe

are dirt. Dirt on the calibration sphere would in principle cause an opposite bias, but in practice the calibration sphere is much cleaner than the hole, and consequently dirt on the calibration sphere has very little effect relative to dirt in the hole.

CONCLUSIONS

We discussed the diameter and geometry of one micro-hole measured using the NIST fiber probe in this paper. The hole was tapered and the diameter increased at the rate of $20.2\text{ }\mu\text{m/mm}$ of depth inside the hole. The radial form showed dominant second, third and fourth order harmonics at the different levels. It does not appear that amplitudes of the different harmonics are consistent at the different depths. We had also measured another hole in the same nozzle, and the diameter of this hole increases at the rate $21.4\text{ }\mu\text{m/mm}$ of depth inside the hole. The hole is predominantly oval and maintains the ovality at different depths. It therefore appears that the different holes, although probably manufactured in similar conditions, show different geometry.

Further, we have also noticed sharp protrusions of about $2\text{ }\mu\text{m}$ to $3\text{ }\mu\text{m}$ in size spanning 2 or 3 sampling points that may be surface features (part form) or could also possibly be dirt or machining debris. If such protrusions are dirt/debris, the uncertainty in diameter will significantly be higher. The expanded uncertainty in diameter ($k = 2$) is 196 nm for most profiles we have measured, but is estimated to be 388 nm when considering protrusions, if present, as dirt/debris.

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PRECISION MEASUREMENT OF PARALLELISM AND ORTHOGONALITY ON MULTI-AXIS MICRO MEASUREMENT SYSTEM

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INTRODUCTION

This study describes about the precision measurement method to evaluate the geometrical accuracy of micro measurement systems, which can measure the micro parts in high accuracy. To realize the micro measurement system, we have to value accuracy of geometrical error of stages such as parallelism between each axis and orthogonal accuracy strictly. In this report, we show new measurement method of evaluating the parallelism between spindle axis (C-axis) and stage axis (z-axis). The principle of this measurement method was already shown in previous study^[1], and just preliminary experiment was performed. In this report, the principle of this method is denoted again simply and has some experiment. The experimental results about this method can evaluate the parallelism of the measurement system. An orthogonal accuracy measurement method between each stage axis is also tested. Because the errors of the right angle in the cube mirror as a standard are cancelled by the summation of the 4-times measurement results, this method does not depend on the accuracy of right angle cube mirror. The basically experiment and analysis is performed and the results guarantee the validity of this method.

MEASUREMENT METHOD FOR THE PARALLELISM BETWEEN SPINDLE AXIS AND STAGE AXIS

FIGURE 1. is a new method to measure the parallelism of z-axis and rotational axis. This method uses a master ball and two displacement sensors. The master ball is attached to the spindle, and the sensor is set to the main axis. The parallelism is evaluated by two z-points measurement. In this method, there

is a miss-alignment between the center of the spindle rotation and the one of the master ball. There is also an attaching position error between two z-points. Therefore, the rotational center Q is measured.

A jig unit holds two displacement sensors and it is able to rotate around the middle point of sensors. This rotation is less than one times. When the displacement sensor measures the master ball in point P, there is only one angle ψ , where P-P' line becomes the perpendicular bisector of O-Q line. At this angle, the outputs of two displacement sensors are same except for the phase. The rotation center exists in the extended line of that middle point. Of course, the rotation center point can be evaluated by one probe scanning on surface, but scanning for the minimum variation of sensor is time consuming. The proposed method is able to detect center point easily just comparing two sensors outputs.

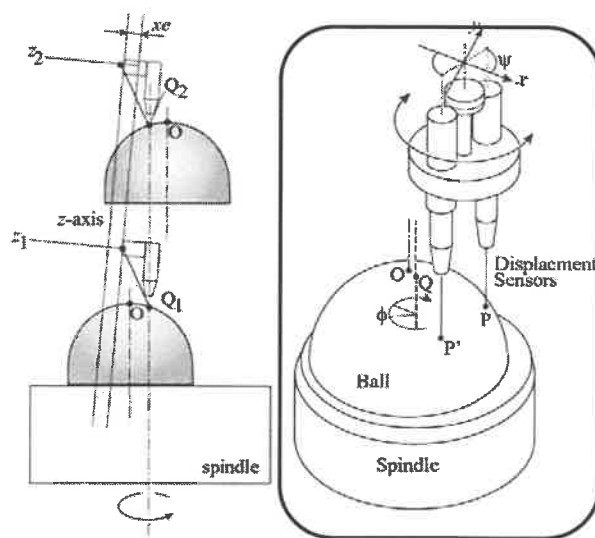


FIGURE 1. Schematic of the measurement method for parallelism evaluation

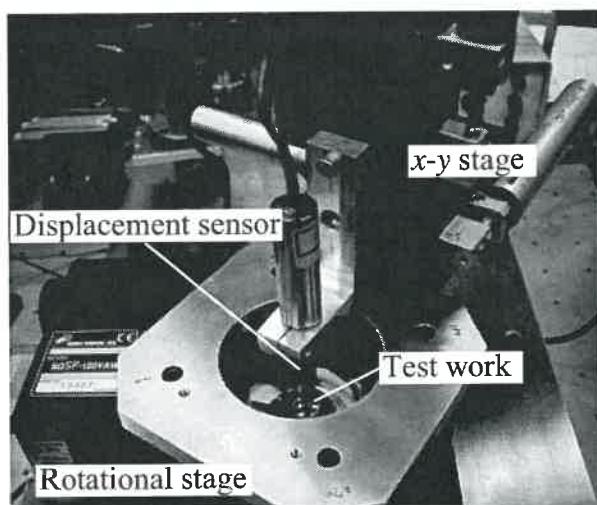


FIGURE 2. System setup for experiment

BASIC EXPERIMENTAL RESULT FOR PARALLELISM MEASUREMENT

FIGURE 2 is an experimental setup for parallelism measurement. One capacitance type displacement sensor (ADE Microsense) is attached to the x-y stage. This stage puts on a θ stage to give a rotational motion in FIGURE 1. Because the x-stage shifts the sensor two times to measure P and P' points, this system is considered to have virtual two probes. The master ball as test work is a steal ball with 40 mm diameter, and the air spindle rotates within 60 rpm in this experiments.

The output in rotational center point is shown in FIGURE 3. P-V amount of this signal is about $0.037 \mu\text{m}$. The most of this output is noise like and 1st order frequency component is almost negligible.

FIGURE 4. shows the variation of differential

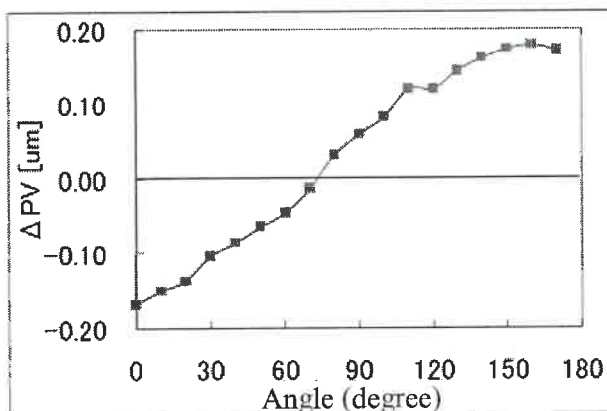


FIGURE 4. Variation of peak-to-valley difference between two sensors' outputs

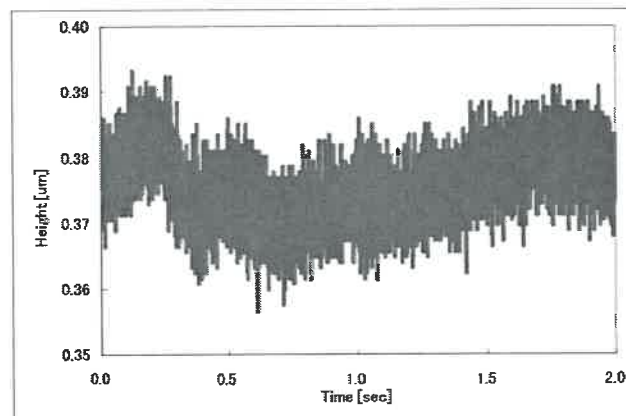


FIGURE 3. Signal output of center position

outputs of two sensors' P-V amount with each rotational stage angle. The point, $\Delta PV=0$, gives a suitable P-P' position which is perpendicular bisector of O-Q line. In this figure, the bisector point is 72 degree.

From this angle, the rotational stage rotates 90 degree and x-stage moves the sensor to detect center position. Typical output of this motion is FIGURE 5. The P-V value decreases with linear relation until 0 to 900 μm and after that, it increases reversely. The output at 900 μm position is like FIGURE 3.

After this measurement, the master ball is attached at Δz higher point. And the system measures the center point of rotational spindle in this height using same procedure as above mentioned. The difference between two measured center points means the parallelism of

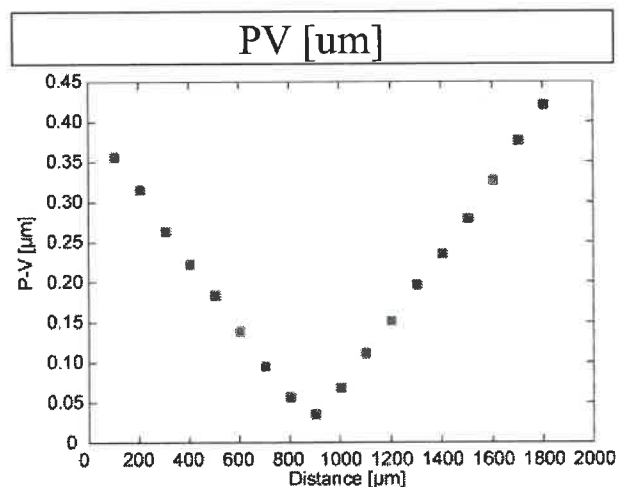


FIGURE 5. Center position detection from $q=162$

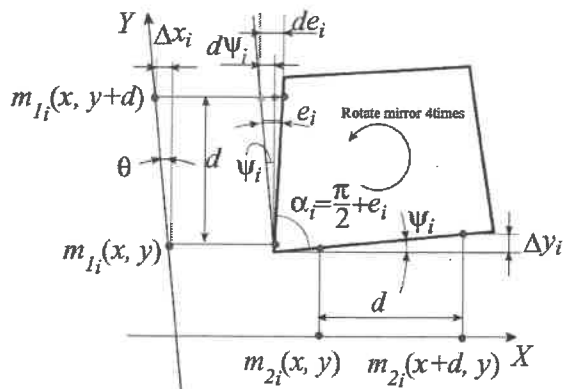


FIGURE 6. Measurement principal for orthogonality evaluation

spindle axis and stage.

Orthogonality measurement of XY stage

Due to many reasons motion in one axis of the given stage cannot be perfectly perpendicular to motion in the other axis. This deviation from perfect orthogonality is called orthogonality error. Measuring amount of that deviation using some noble algorithm and device is main theme of this research.

Theoretical principle of this measurement of orthogonality of XY stage is in FIGURE 6.

θ is orthogonality error, e_i is mirror profile error from right angle and ψ_i is setting error. The relationship between these values are as follows:

$$\theta = \frac{m_{1i}(x, y+d) - m_{1i}(x, y)}{d} + e_i + \psi_i$$

$$= \frac{\Delta m_{1i}}{d} + e_i + \psi_i$$

Because of geometrical theory, summation of 4 corners angles becomes obviously 2π . Then,

$$\sum_{i=1}^4 \alpha_i = \sum_{i=1}^4 \frac{\pi}{2} + e_i = 2\pi + \sum_{i=1}^4 e_i$$



FIGURE 7. System set-up for orthogonal measurement

$$\therefore \sum_{i=1}^4 e_i = 0$$

The profile errors are canceled. Finally, orthogonality error is calculated as below:

$$\theta = \frac{1}{4} \sum_{i=1}^4 \left(\frac{\Delta m_{1i}}{d} + \psi_i \right)$$

Almost same method was already mentioned and evaluated theoretical uncertainty in previous study^{[2], [3]}. But there are few experimental results. Then this time, we had a series of experiment to confirm this method in experimentally.

System setup and experimental results

Device set-up for stage control and measurement experiment is as FIGURE 7. XY stage (Suruga Seiki K101-30LHD-M) system is used for measured stage. The laser encoder (Renishaw RLD10-3P) measures the motion of x-y stage. The cube mirror and rotational stage is put on the top of the x-y stage to measure the orthogonality error.

The measurement experiment is conducted. For guarantee of repeatability of this measurement method, these below experiment conditions are

TABLE 1. Results of orthogonality measurement

Angle	$\theta + e$ (rad)	
	0 deg	1 deg
1	0.0035552	0.0035643
2	0.0039835	0.0035395
3	0.0035425	0.0040005
4	0.0040035	0.0040011
Orthogonality error	0.0037712 rad 0.2160800 deg	0.0037764 rad 0.2163760 deg

used. 1) 4 time experiments are conducted. 2) For each time, stage including mirror and laser interferometer are reset at arbitrary position. 3) For each time, moving distance is 20 mm which is restricted by size of cube mirror and motion velocity is set to 20000 pps, or 400 $\mu\text{m/s}$. 4) 1 more time experiment of different moving distance, 10 mm is additionally conducted.

The result of experiment is shown in TABLE 1.

The orthogonality error measured in both experiment has deviation of about $9.00\text{e-}7$ rad or $4.75\text{e-}5$ deg. corresponding to about 18 nm linear difference when stage moves 20 mm. Deviation of orthogonality error is very small and obtained result is reasonable. Through the results above, it is concluded that orthogonality error obtained by method suggested here has reliability in spite of some uncertainties.

Conclusion

This paper is summarized as follows.

- 1) New measurement method of evaluating the parallelism between spindle axis (C-axis) and stage axis (z-axis) is proposed. The

experimental results about this method evaluate the parallelism of the measurement system.

- 2) An orthogonal accuracy measurement method between each stage axis is also tested. From basic experiment and consideration, the orthogonality errors can be obtained and these results are reasonable.

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CORRECTED MEASURED POINT VALIDITY CHECK IN COORDINATE METROLOGY

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INTRODUCTION

One of the most difficult measurement tasks is to scan an unknown surface with a geometrically complex shape. The center coordinates of the previously calibrated stylus tips are recorded by the coordinate measuring machine and must be corrected to provide an estimate of the actual point measured on the surface of the object.

CMMs typically have one or more built-in algorithms to calculate corrected measured points. However, the algorithms often do not allow proper estimation of corrected measured points, especially when measuring the shape of surfaces with highly varying curvature [1] and do not permit the self-assessment of validity for the points measured. Some attempts to estimate uncertainty using the so-called virtual CMM which were initiated [2] but have not yet been widely used in commercial equipment.

A new algorithm for stylus tip radius correction enables not only to calculate corrected measured points much more precisely, but also to assess the validity of individual measured points. It provides CMM user with a tool to:

- evaluate the correctness of measurements;
- optimize the measuring process by selecting an appropriate configuration of stylus tips, measuring speed, and sampling step;
- eliminate certain points of the measured outline if for any reason they have been measured incorrectly.

STYLUS TIP RADIUS CORRECTION

Corrected measured points (estimators of the actual point of tangency between the stylus tip and the surface of the measured object) are estimated by adding a vector, of length equal to the effective stylus tip radius, to the indicated points [3]. The direction of this vector should be

the direction normal to the surface of the measured object at the point of tangency with the stylus tip.

Many solutions are known that can be used to determine the direction of the stylus tip radius correction vector making use of the information on the reciprocal position of neighboring indicated measured points [4], [5], [6], [7], [8], [9].

The correction vector can be estimated as the perpendicular to the straight line running through the indicated point and the previous or next point. Due to a high density of measured points, minor errors in their positioning may generate serious errors in the positioning of corrected points. Such a phenomenon is particularly visible in measurements of surfaces with rapidly changing curvature.

A better method is to use NURBS curves, which are used to approximate the path of the stylus tip center. Then the direction of the correction vector is determined as perpendicular to the curve. This reduces the number of errors caused by the wrong direction of the correction vector, but leads to the smoothing of the actual shape of the surface, and therefore points may be generated in places inaccessible to the stylus tip due to the constraints of its diameter. Such an effect can be noted in measurements of concave surfaces.

NEW CORRECTION ALGORITHM

A new algorithm for corrected point calculation was proposed [10] that does not require a correction vector to be calculated. The algorithm is intended for high-density sampling, i.e. those in which the distance between indicated measured points is considerably smaller than the diameter of the stylus tip.

CORRECTED POINT POSITION VALIDITY CHECK ALGORITHM

position of a corrected measured points results from errors in the measurement of indicated measured points and errors in the determination of the length of the effective radius of the stylus tip.

Such an assessment can be done for each indicated point O_i by checking a simple condition defining the validity parameter called validity check (VC):

$$\begin{aligned} |A_{i+1} - O_{i-1}| < R &\xrightarrow{\text{def}} VC = 0 \\ |A_{i+1} - O_{i-1}| \geq R &\xrightarrow{\text{def}} VC = 1 \end{aligned} \quad (1)$$

The algorithm described using equation (1) can be modified by allowing for the cases in which points are located within a certain interval of uncertainty which according to the definition of uncertainty can be assigned to the measurement outcome. Such an interval can be illustrated by means of a ring of resultant uncertainty width u_P , concentric with the stylus tip's trace at indicated point O_{i-1} as depicted in FIGURE 2.

$$u_0 = THN / (2\sqrt{3}) \quad (2)$$

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between the results achieved in tests on a certified test sphere. Uncertainty u_R can be calculated as standard deviation s_R of the radial distances between the measured points of the reference sphere used for the qualification.

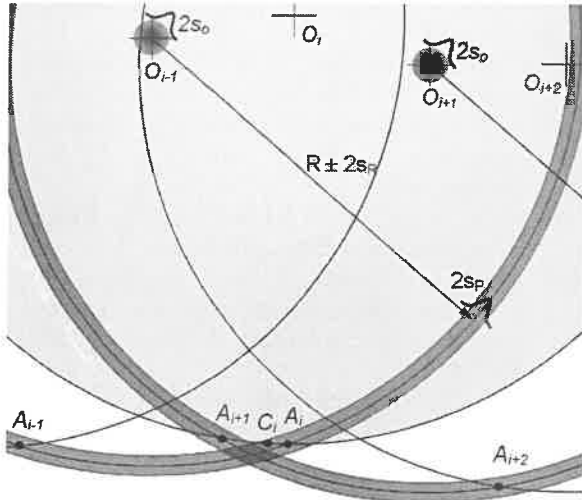


FIGURE 2. Geometrical analysis of validity for the position of corrected measured points taking into account the uncertainty of measurement.

In such case, because the error is Gaussian, the probability that corrected measured point C_i is not within the material stylus tip when measuring any of the indicated points can be calculated as the lowest value of probability $P_{i,n}$ calculated for each of $n \in \{1, 2, \dots, i-1, i+1, \dots, N\}$ points separately,

$$P_{i,n} = \frac{1}{\sigma\sqrt{2\pi}} \int_{-\infty}^{R_{i,n}} \exp\left[-\frac{(R_{i,n} - R)^2}{2\sigma^2}\right] dR_{i,n}, \quad (3)$$

where, N stands for the number of measured points, $R_{i,n}$ relates to the distance between corrected point C_i and indicated point O_i :

$$R_{i,n} = |C_i - O_n| \quad (4)$$

and

$$\sigma = s_P = \sqrt{\left(\frac{THN}{2\sqrt{3}}\right)^2 + s_R^2} \quad (5)$$

However, calculating probability $P_{i,n}$ makes sense only when $R_{i,n}$ is not higher than R .

The probability $P_i = \min(P_{i,n})$ defines the validity of corrected measured point C_i .

EXPERIMENTAL TESTS

Measurements of a section of a leak stopper groove in a combustion engine head were carried out on a Zeiss ACCURA CMM fitted with an active VAST Gold scanning probe. The maximum permissible error of indication for size measurement of the ACCURA amounts to $MPE_E = 1.7 + L/333 \mu\text{m}$, where L stands for the length measured in millimeters. The maximum permissible scanning probing error MPE_{Tij} amounts to $2.7 \mu\text{m}$. The item was measured with a 1.5 mm stylus ball diameter and a measuring speed of 1 mm/s. The sampling step was 0.05 mm. FIGURE 3 shows the measurement results as two sets of points. The first set, continuous line and full circle markers, is a set of indicated measured points (ball centers) O_i . The second set, identified with a dashed line and squares, are the corrected measured points C_i obtained using the correction process.

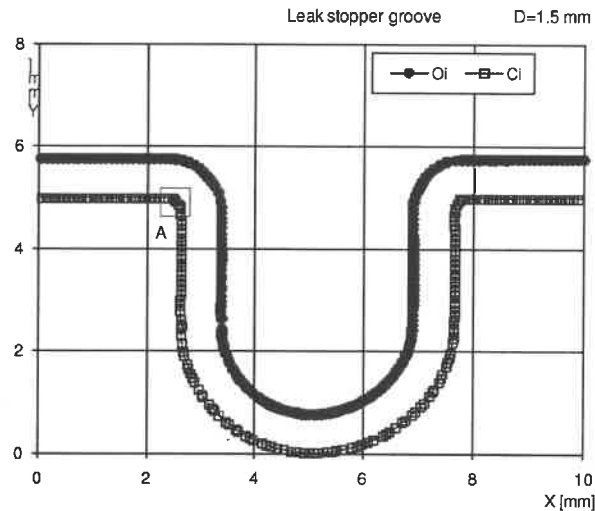


FIGURE 3. The result of measuring the section of a leak stopper groove using an ACCURA.

A section corresponding to an edge is presented separately as detail A in FIGURE 4. For each measured point, a validity value is calculated by determining probability P_i . It is a quite troublesome case of scanning measurements, which is well known to coordinate measuring machine users. Minor errors in the positioning of indicated measured points occurring when measuring such shapes e.g. as a result of the imperfections of probe transducers or the coordinate measuring machine itself, make the calculation of the direction of the correction vector evidently incorrect. This phenomenon is augmented by the fact that in measurements of precise elements with a high density of

measured points the distance between subsequent indicated measured points is much smaller than the stylus tip radius. Consequently, erroneous corrected measured points result, often incorrectly ordered, which was demonstrated in previous publications [1], [10]. In the case shown in FIGURE 4 the correction algorithm has properly calculated corrected points. There are no visible deformations of the measured profile. However, parameter P_i indicates that the points measured on the edge are less valid. This provides the coordinate measuring machine user with a tool to accept or reject such measured points.

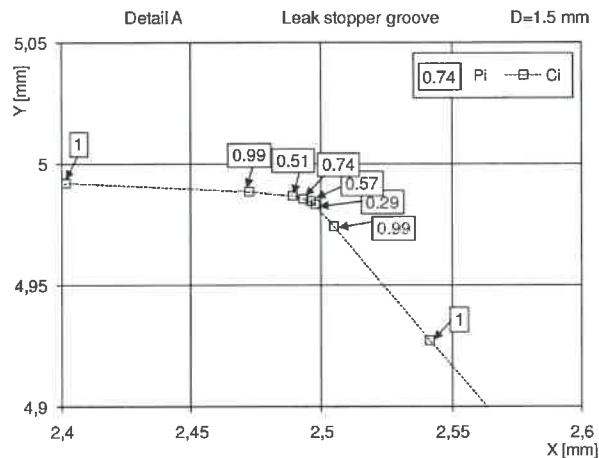


FIGURE 4. Result of measuring the section of the groove – detail A.

SUMMARY

The paper presents a new algorithm for check of the validity of measured points in scanning measurements carried out on coordinate measuring machines. It makes it possible for users to evaluate the correctness of measurements, and justify their elimination. Monitoring of probability P_i for individual measured points or their mean enables to optimize the measuring process by choosing an appropriate configuration of stylus tips, measuring speed, and sampling step to achieve the best possible reliability of measured points.

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DEVELOPMENT OF MICRO MECHANICAL PARTS UTILIZED EXTRA FINE METALLIC WIRES OR CERAMIC THREAD

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INSTRUCTIONS

Research and development of the micro machine which is called MEMS (Micro Electro Mechanical System) have been done with the big expectations of many people, since a birth of the micro machine in the 1980s. In the big expectations toward the micro machine, many people have been waiting for appearing of a new machine which had not been gained in the history of man. Under the big expectations of the people, researchers and engineer developed three major methods of photolithography[1], LIGA process[2], and ultra-precise milling[3]. They are produced the micro mechanical parts such as a micro gear, a micro linkage, and a micro motor. However, if we choose these methods to make the micro mechanical parts, we have to install huge and expensive facilities. Moreover, if the micro mechanical parts are designed in three dimensionally complex shapes, we can not manufacture by these methods. Therefore, many researchers and engineers wait for the forth method to developed micro mechanical parts with small facilities, simple process, and high making ability, without expensive equipments. A novel method to make the micro mechanical parts utilized extra fine wires was developed, so that a micro screw, a micro nut, a micro helical gear, and a micro turbine impeller were manufactured by the developed method[4].

In this paper, a micro spiral bevel gear utilized extra fine metallic wires is developed by the advanced manufacturing method, and the meshing theory with the micro spiral bevel gears utilized extra fine metallic wires is analyzed theoretically. Additionally, the ceramic screw and ceramic nut are manufactured as applications by the advanced manufacturing method.

THE MICRO SPIRAL BEVEL GEARS UTILIZED EXTRA FINE METALLIC WIRES

The micro spiral bevel gears utilized extra fine metallic wires are shown in Figure 1. The micro spiral bevel gear is made by brazing the

extra fine metallic wires which are arranged around a small cone. The micro spiral bevel gears have dimensions as shown in Table 1.

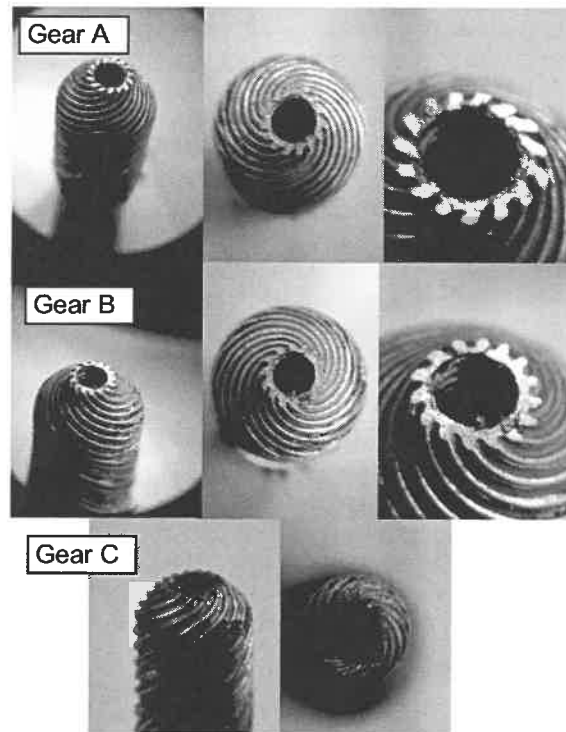


FIGURE 1. The micro spiral bevel gears utilized extra fine metallic wires.

TABLE 1. Dimensions of the micro spiral gears utilized extra fine metallic wires.

	Gear(A/B)	Gear(C)
Module	255 μm	127 μm
Diameter of the extra fine metallic wires	400 μm	200 μm
Number of Teeth	15	15
Direction of Spiral	R /L	R
Pitch Diameter	10.4 mm	4.2 mm
Outside diameter	10.8 mm	4.4 mm
Face Width	4.3 mm	1.4 mm
Pitch angle	45 deg	45 deg

MAKING PROCESS OF THE MICRO SPIRAL BEVEL GEAR UTILIZED EXTRA FINE METARIC WIRES

The micro spiral bevel gear utilized extra fine wires is made through following processes as shown in Figure 2.

- (1) A small axis which top is shaped into a cone and two type's wires of extra fine metallic wires are prepared. One type's wires are firmly soldered and another type's wires are hardly soldered.
- (2) Copper film is covered on the conical top and side of the small axis.
- (3) The small axis is inserted into the center hole of the winding apparatus which has a rotating disc with many pin holes at its rim.
- (4) A center pole is attached at the conical top of the small axis, and two type's wires are fixed on the center pole. Two type's wires are alternately inserted into the pin holes.
- (5) Two type's wires are bound by a binding wire from tip to base of the center pole.
- (6) By rotating the disc of the winding apparatus, two type's wires are closely and alternately arranged on the cone of the small axis. Two type's wires shape into the conical spiral.
- (7) Melted solder is penetrated into the gap between the two type's wires and the copper film. The extra fine wire to be firmly soldered is brazed on the small cone, and yet the extra fine wire to be hardly soldered is not brazed on the small cone.
- (8) The wires to be hardly soldered are removed.
- (9) The extra fine wires to be firmly soldered are remained as teeth of the micro spiral bevel gear.

MESHING THEORY WITH THE MICRO SPIRAL BEVEL GEARS UTILIZED EXTRA FINE METARIC WIRES

The meshing theory with the micro spiral bevel gears utilized extra fine metallic wires is analyzed by using the geometrical conical models as shown in Figure 3. As the micro spiral bevel gear consists of a truncated cone and extra fine wires; also, the wires are soldered on the conical side of the truncated cone with the same teeth's width and with the same ditch's width from the center to the periphery of the spiral bevel gear, the meshing theory is built by analyzing geometrical location of the wires on the fan-shaped sheet in which the conical side spread.

Figure 4(A) shows the wires location on the circular sheet with a center hole. The two type's

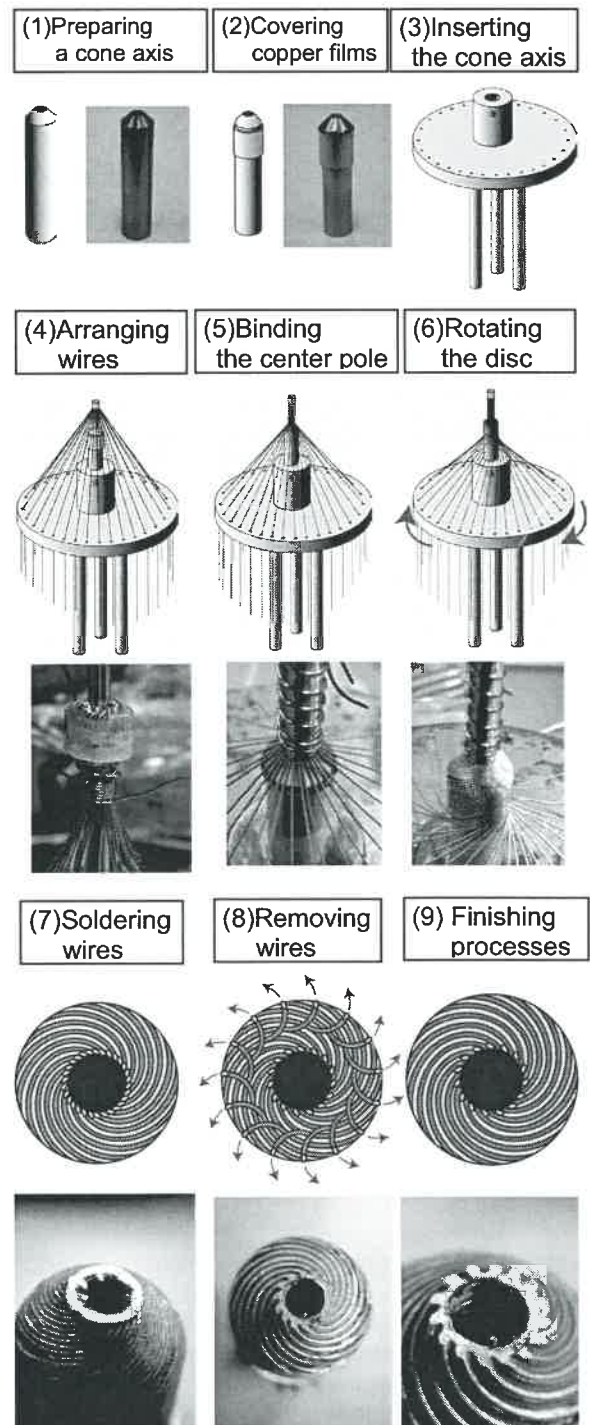


FIGURE 2. Making process of the micro spiral bevel gear utilized extra fine metallic wires.

wires are closely and alternately arranged toward CCW on the circular sheet, and one type's wires are soldered on the circular sheet and another type's wires are removed from the

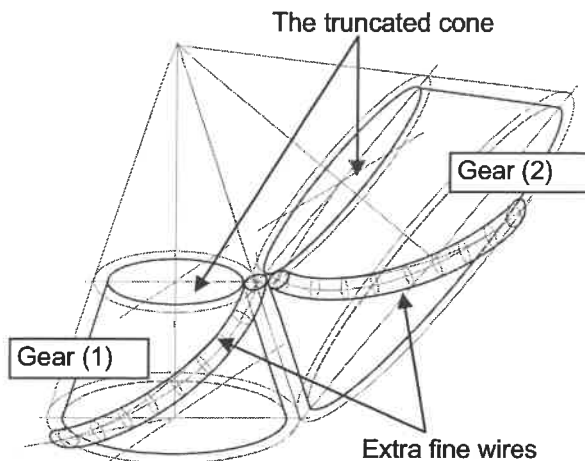


FIGURE 3. Analytical model for meshing theory of the micro spiral bevel gear utilized extra fine metallic wires.

circular sheet. In the Figure 4(A), solid lines indicate the teeth ridges, and broken lines indicate the teeth ditch. Both lines become involutes' curves. When the circular sheet is cut in a fan-shaped sheet with a certain center angle as shown Figure 4(B) and the fan-shaped sheet is assembled into a truncated cone, the truncated cone composes a spiral bevel gear (i.e. Gear (1)) utilized extra fine wires.

On the other hand, two type's wires are closely and alternately arranged toward CW and one type's wires are soldered on the circular sheet and another type's wires are removed from the circular sheet as shown Figure 4(C). When the circular sheet is cut in a fan-shaped sheet with a certain center angle as shown Figure 4(D) and the fan-shaped sheet is assembled into a truncated cone, the truncated cone composes a spiral bevel gear (i.e. Gear (2)) utilized extra fine wires.

As the both circular sheet in which the wires are wound into the spiral shape toward CCW or CW have the relation of the front and back with the same dimensions and the same inclines of the wires, even if the center angles of the fan-shaped sheets are different, it is possible to mesh with two spiral bevel gears each other.

Figure 5 shows the meshing condition with the spiral bevel gears utilized extra fine wires. We could manually rotate the gears.

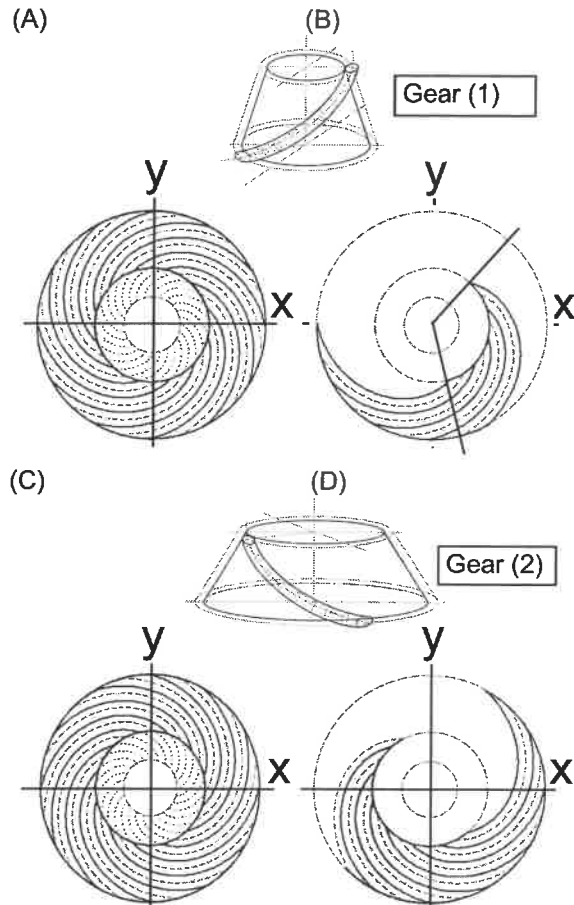


FIGURE 4. Circular sheets and fan-shaped sheets composed the spiral bevel gears.

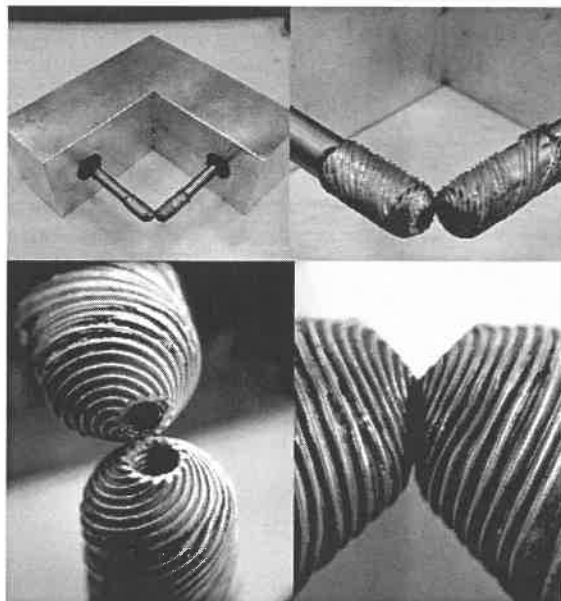


FIGURE 5. Meshing Experiment with the micro spiral bevel gears.

THE CERAMIC SCREW AND THE CERAMIC NUT UTILIZED CONTINUOUS ALUMINA-SILICA THREADS

Figure 6 shows a micro screw and a micro nut utilized continuous alumina-silica threads. The micro-screw and the micro-nut have a pitch of 0.32 millimeters, and their major diameters are 6.32 millimeters and their lengths of threaded portion are 3.5 millimeters. The ceramic screw and the ceramic nut are manufactured by gluing the continuous alumina-silica threads to an alumina-silica stick or an alumina-silica tube with ceramics paste.

The ceramic micro screw is made through following processes. (1) An alumina-silica stick is prepared as shown in Figure 7-(S1). (S2) A continuous alumina-silica thread and a copper wire are fixed on the alumina-silica stick. (S3) Ceramics paste is applied. (S4) The continuous alumina-silica thread and the copper wire are closely coiled each other around the alumina-silica stick. (S5) Surplus ceramics paste is wiped out. (S6) Ceramics paste is sintered. (S7) The copper wire is melted in a solution of nitric acid. (S8) The ceramic micro screw is produced.

The ceramic micro nut is made through following processes. (N1) A Copper film is covered on the alumina-silica stick as shown in Figure 7-(N1). (N2) A continuous alumina-silica thread and a copper wire are fixed on the alumina-silica stick. (N3) Ceramics paste is applied on. (N4) Ceramics paste is sintered. (N5) The alumina-silica stick is pulled out. (N6) The copper wire and the copper film is melted in a solution of nitric acid. (N7) The ceramic micro nut is produced.

CONCLUSIONS

- (1) The novel method to make a micro spiral bevel gear was developed. The micro spiral bevel gear was produced by brazing extra fine wires onto the conical axis.
- (2) The meshing theory was built by analyzing geometrical locations of the wires on the fan-shaped sheet spread the conical side.
- (3) The ceramic screw and the ceramic nut were manufactured as applications by the advanced manufacturing method.

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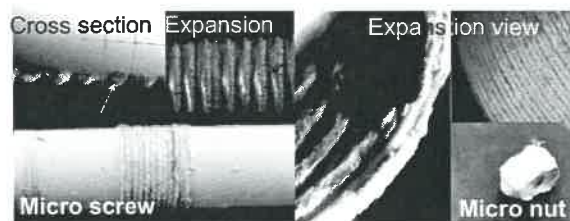


FIGURE 6. The ceramic screw and the ceramic nut utilized continuous alumina-silica thread.

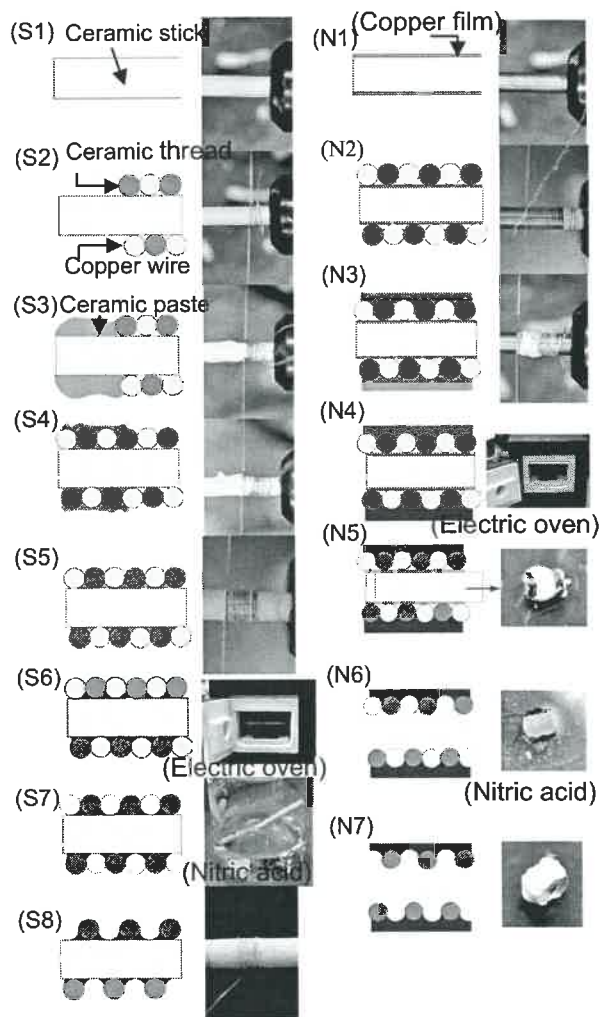


FIGURE 7. Making process for the ceramic screw and the ceramic nut utilized continuous alumina-silica thread.

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FABRICATION OF A MICROROBOT OBSERVABLE IN AN ARTIFICIAL HUMAN'S LARGE INTESTINE MODEL

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INTRODUCTION

An inspection of the large intestine is very effective in order to prevent the large intestine cancer. A fiberscope is excellent for the inspection of the large intestine. However, the inspection by the fiberscope often attends many pains and injuring of the large intestine. We are hoping the inspecting method for the large intestine with no pain and no injuring of the large intestine before the operation by the fiberscope. Many microrobots that can move in the large intestine have been proposed by several research groups [1], [2]. However, the sure moving in the large intestine has not been confirmed, because the actual large intestine in our human body is flexible and has very severe bending curve at the sigmoid colon, the splenic flexure and hepatic flexure. Then the mobile moving in the large intestine is very difficult.

Now, we propose a new microrobot that can surely move in the large intestine in the human body. We need a holding work to the radius direction of the flexible large intestine and a driving work to longitudinal direction of the large intestine using the friction force by the holding of the large intestine, in order to move surely in the flexible and severely curved large intestine, for example, that is a sigmoid colon. We use seven rubber bellows in series. Two outer rubber bellows are supplied six bulging rubber sheets. These are called as the braking mechanism. The braking mechanism works as a brake of the microrobot. The second rubber bellows from the front rubber bellows is called a moving mechanism. The moving mechanism moves the microrobot by stretching and shrinking. Four rubber bellows are arranged after the moving mechanism in a matrix condition. These are called as the bending mechanism. The bending

mechanism can bend the microrobot more than 100 degrees by its bending motion. The microrobot can pass the sigmoid colon. When the rubber bellows of the braking mechanism shrinks, six bulging rubber sheets spread to the radius direction and touch the large intestine. Then the spread bulging rubber sheets surely hold the large intestine. When the rubber bellows of the braking mechanism stretches, spreading to the radius direction of the six bulging rubber sheets and touching to the large intestine are canceled. In this condition, the microrobot can be inserted into the anus. The rubber bellows are stretched by pneumatic pressure and shrank by vacuum pressure. The stretching, shrinking and bending motion of the rubber bellows are controlled by a computer and electromagnetic valves.

The microrobot has been confirmed to move and observe from the anus to the transverse colon through the sigmoid colon in the artificial human's large intestine model for the freshman doctor's inspection training using the fiberscope.

ARTIFICIAL HUMAN'S LARGE INTESTINE MODEL

An artificial human's large intestine model is shown in Figure 1. The numbers in the figure are the diameter of the large intestine model. The large intestine model is made of rubber. We see the anus downward of the figure. There exists the rectum following the anus. The sigmoid colon, the descending colon, transverse colon and the ascending colon follow the rectum. The cecum follows the colons. The large intestine consists of the rectum, the sigmoid colon, the descending colon, transverse colon, the ascending colon and the cecum.

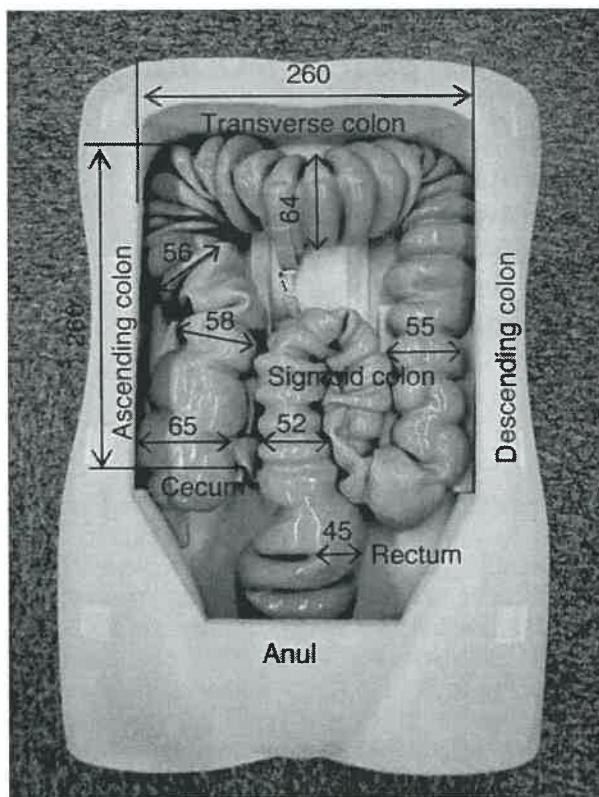


FIGURE 1 An artificial human's large intestine model.

STRUCTURE OF THE FABRICATED MICROROBOT

Structure of the fabricated microrobot is shown in Figure 2. The microrobot consists of two braking mechanisms, a driving mechanism and a bending mechanism, LED for the light and a CCD camera. The two braking mechanisms and driving mechanism are driven by bellows which are 16 mm in outer diameter, 35 mm long and made of nitrile butyl rubber (NBR). The bending mechanism is driven by four bellows which are arranged as matrix and are 8 mm in outer diameter, 61 mm long and made of NBR. Each bellows is an independent vessel. Seven air feeding tubes that are 1.25 mm in outer diameter, 0.65 mm in inner diameter and 1 m long are connected to the two braking mechanisms, the moving mechanism and the bending mechanism. The bulging rubber sheets for two braking mechanisms are 7 mm wide, 2 mm thick, 36 mm long and made of NBR. The bulging rubber sheets are reinforced by plastic sheets. The rubber bellows are stretched by pneumatic pressure (positive pressure) and shrank by vacuum pressure (negative pressure).

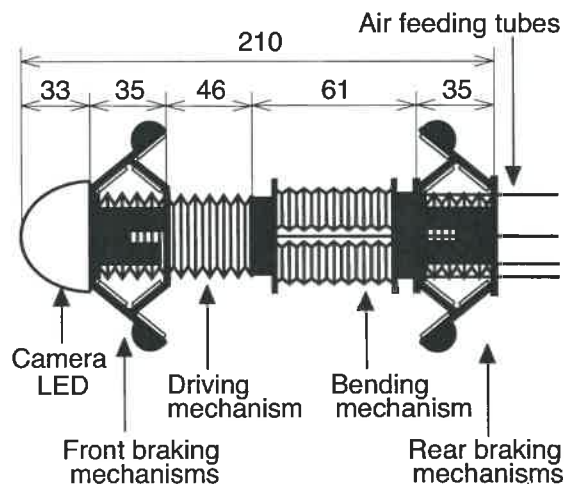


FIGURE 2 Structure of the fabricated microrobot.

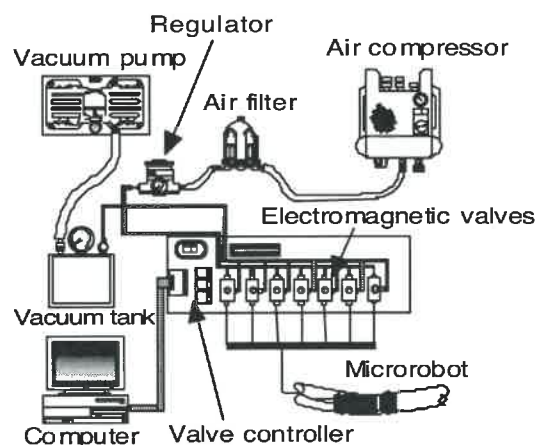


FIGURE 3 Experimental apparatus.

The stretching, shrinking and bending of the rubber bellows are controlled by a computer and electromagnetic valves.

EXPERIMENTAL APPARATUS

An experimental apparatus for measuring the characteristics of the microrobot is shown in Figure 3. A computer controls seven electromagnetic valves through a valve controller. Seven air feeding tubes are connected from the electromagnetic valves to the two braking mechanisms, the driving mechanism and the bending mechanism of the microrobot. An air compressor is connected to the entrance port of the electromagnetic valves and feeds the positive pressure in order to stretch the bellows. A vacuum pump is

connected to the another port of the electromagnetic valves and feeds the negative pressure in order to shrink the bellows.

MOVING PRINCIPLE OF THE MICRORBOT AT THE SIGMOID COLON

The moving principle of the microrobot at the sigmoid colon is shown in Figure 4.

Step 1: At initial, all the bellows are shrinking by the negative pressure. The bulging sheets of the front and rear braking mechanisms are at the condition of the braking and hold the rectum.

Step 2: The positive pressure is supplied to the front braking mechanism and the braking is canceled.

Step 3: The positive pressure is supplied to the moving mechanism and the two outer bellows of the bending mechanism. The bending mechanism bends more than 100 degrees to inside direction, because the two outer bellows stretch. Then, the driving mechanism can move into the sigmoid colon.

Step 4: The negative pressure is supplied to the front braking mechanism. The front braking mechanism is in the condition of the braking and holds the sigmoid colon. At that time, the positive pressure is supplied to the rear braking mechanism and the braking is canceled.

Step 5: The negative pressure is supplied to the driving mechanism and the two outer bellows of the bending mechanism. The rear part of the microrobot can move to the front direction, because bellows of the driving mechanism and the two outer bellows of the bending mechanism shrink.

Step 6: The negative pressure is supplied to the rear braking mechanism. The rear braking mechanism is in the condition of the braking and holds the sigmoid colon. It is same as the initial condition and one cycle is over. The microrobot can pass the sigmoid colon and the splenic flexure by this series of the step.

EXPERIMENT

Observation in the Large Intestine

A photograph of the inside of the large intestine observed by the CCD camera of the microrobot is shown in Fig. 5

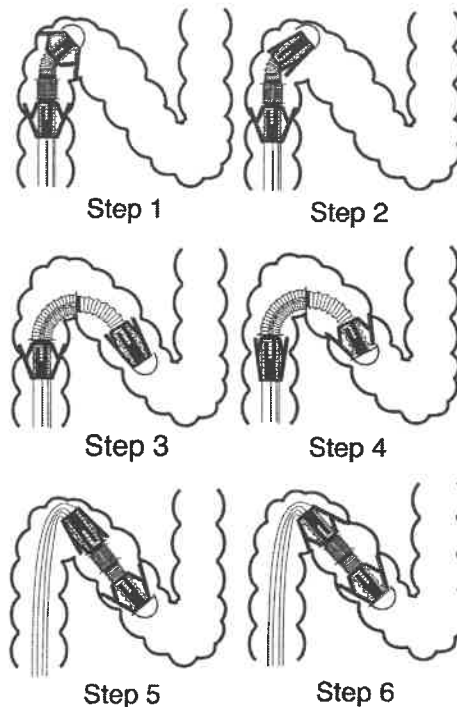


FIGURE 4 The moving principle of the microrobot at the sigmoid colon.

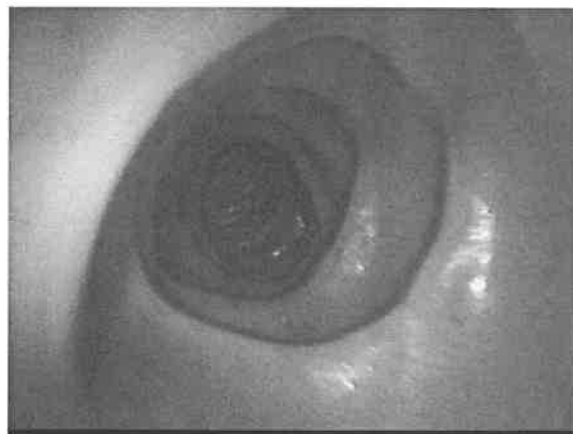


FIGURE 5 A photograph of the inside of the large intestine observed by the CCD camera of the microrobot.

Characteristics of the Bending Mechanism

Characteristics of the bending mechanism are measured. First, the negative pressure of -80 kPa is given to all the bellows of the bending mechanism. The angle of the bending mechanism is 0 degree. Then, two left bellows are given the positive pressure. The bending

mechanism bends to right direction. Similarly, two right bellows are given the positive pressure. The bending mechanism bends to left direction. The relationship between the positive pressure and the bending angle is shown in Figure 6. In the case of +80 kPa of the positive pressure, the bending angle of 140 degrees is obtained on the right bending action and the bending angle of 139 degrees is obtained on the left bending action.

Moving Experiment

Silicon oil is splayed inner wall of the large intestine model. The microrobot is inserted into the anus of the large intestine model. The microrobot can move to the hepatic flexure through the rectum, the sigmoid colon, the descending colon and the transverse colon.

Traction force and speed of the microrobot are measured. The relationship between the traction force and the speed is shown in Figure 7. The microrobot can move at the speed of 3.7 mm/s in the traction force of 0.3 N.

CONCLUSION

(1) We fabricated a new microrobot that can move in an artificial human's large intestine model for the freshman doctor's inspection training using the fiberscope. The microrobot consists of two braking mechanism, a moving mechanism, a bending mechanism, LED for the light and a CCD camera. The braking mechanism has a bulging brake which consists of six bulging rubber sheets.

(2) We confirmed by the experiment that the fabricated microrobot can move to the hepatic flexure through the rectum, the sigmoid colon, the descending colon and the transverse colon.

(3) The microrobot can move at the speed of 3.7 mm/s in the traction force of 0.3 N.

(4) This microrobot may be used to the inspection mobile microrobot for the human's large intestine, because the artificial human's large intestine model is very similar to the human's large intestine.

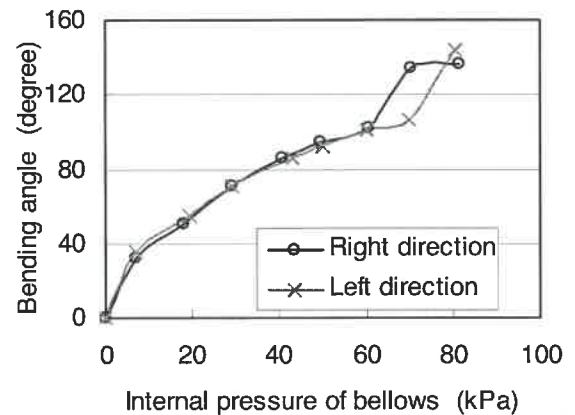


FIGURE 6 The relationship between the internal pressure and the bending angle.

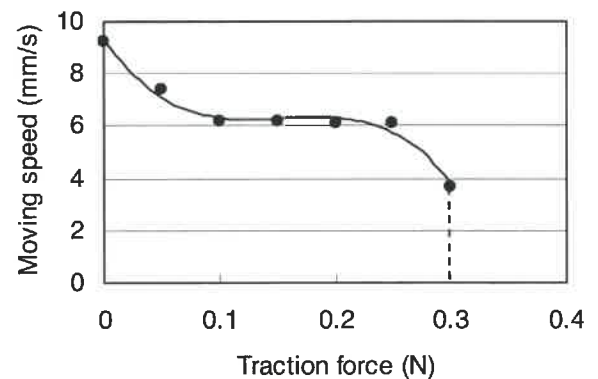


FIGURE 7 The relationship between the traction force and the moving speed.

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MACROSCOPIC TRANSFER ARTIFACT FOR NANO-POSITIONING

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ABSTRACT

Nano-positioning and measuring instruments are the key components that led to micro-fabrication miniaturization. The Sub-Atomic Measuring Machine (SAMM) at UNC Charlotte is a machine that can position very accurately [1]. MIT is developing an atomic force microscope (AFM) for the SAMM stage giving the capability of not only positioning but measuring [2]. It is necessary to correct for the XY mapping errors present in the SAMM stage. Also, it is desired to determine the uncertainty of the apparatus. In order to do so, a transfer artifact must be developed and measured on other machines. This paper describes the preliminary design of an artifact that could be employed to correct for the errors of SAMM and estimate the uncertainty of some nano-positioning instruments.

INTRODUCTION

Demands for viable tools that are capable of measuring and positioning with sub-atomic resolution led to the development of the Sub-Atomic Measuring Machine (SAMM) in the late ninety's. The SAMM stage has a resolution of 0.01nm and a scanning range of 25mm X 25mm. This stage was designed at UNC Charlotte by M. Holmes for his Doctoral thesis [1] and is currently being updated [3, 4]. The specified resolution has been obtained, however the uncertainty is still unknown.

To reach the target uncertainty, error mapping techniques are necessary. Also, in order to check the uncertainty estimate of the machine, an international inter-comparison is desirable with different machines measuring the same artifact.

We are designing and fabricating an artifact with preferentially etched pits that will be used with an algorithm to correct for the x and y errors in the machine by using a mapping procedure [5] and to estimate the measurement. This standard will be fabricated from silicon due to its mechanical properties, presumed temporal

stability, cost, ease of machinability and knowledge of the preferential etching process.

The mounting procedure of the substrate onto the stage is critical. A three point kinematic support system will be employed on the back surface of the standard to control the bending due to gravity. Guidelines published by Loewen [6] will be used to determine the optimal placement of the supports.

The features on the substrate include 4000 nm truncated pyramids etched into the [100] plane with a pitch of 1mm and narrow grooves in between. The location of the centers of these pits will be identified using a high accuracy atomic force microscope (AFM). Currently, an AFM is being developed by MIT to integrate with the SAMM stage [2].

The etching of the silicon and the procedure to locate the pits will be studied first. Other issues will be examined later, such as the flatness of the substrate, purity and cutting of the silicon, cleanliness, and the shipping method. The artifact will be employed on the SAMM stage to correct for systematic errors, and it will also be measured in other laboratories to determine the uncertainty of nano-measuring instruments.

BACKGROUND

XY Mapping Error

The errors present in the SAMM stage are the differences between the real and command positions. It is necessary to correct for these errors in order to improve the performance of the system. An XY error mapping technique will be used with the artifact being developed. Error mapping is a method to quantify the repeatable instrument errors inherent and to separate them from the geometric errors of the artifact. The locations of the pits on the artifact must be measured in 3 positions, normal, rotated by 90° and shifted by 1mm, [5]. It is required that the metric be established separately.

Preferential Etching

Silicon has four covalent bonds that are arranged tetrahedrally which is called the diamond-cubic structure. Silicon wafers are categorized into three orientations, [100], [110], and [111] [7].

Preferentially etching is etching occurring along these selected crystallographic planes. The geometry and size of the pit is dependent on the mask features and wafer orientation. [100] oriented silicon etches anisotropically at an angle of 54.74° .

To achieve a truncated pyramid or v-groove, the mask must have a square or rectangular opening. Figure 1 displays the etching geometry of (100) silicon.

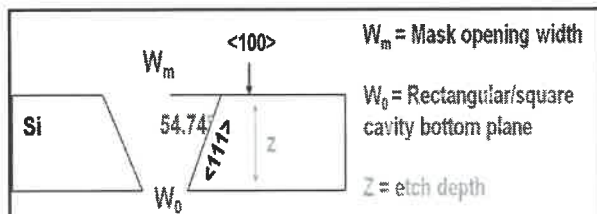


FIGURE 1. Comparison of mask opening width to cavity bottom plane

The etch depth is related to the mask opening width. Using trigonometry and assuming that the cavity bottom is zero, Equation 4 shows the calculation of the depth.

Equation 4.

$$z = \frac{W_0}{\sqrt{2}}$$

CURRENT DESIGN

As stated previously, single-crystal silicon will be used due to its mechanical properties and etching capabilities. It is desired to acquire a substrate that is ideally flat and "stress-free" material. Intrinsic silicon will be used to prevent any stresses due to crystal dislocations in the material resulting from the doping of silicon. The quality of intrinsic material will be carefully examined before the decision is made on where to obtain the material.

Increasing the thickness of the artifact will decrease the wafer deformation. The maximum change in position of the features on the substrate due to deflections was calculated

using Loewen [6]. The values of elastic modulus, density, and Poisson's ratio of silicon used were 2.32 kg/m³, 129.5 GPa, and 0.22, respectively [7].

At a thickness of 0.25mm, the cosine error due to bending is 0.136 pm, which is less than our goal. The thickness chosen was 1mm, which has a displacement of nearly zero. The artifact will include 4000 nm etched pits etched arrayed at a 1mm pitch. There will be 10000nm wide grooves in between the pits for a broad visible placement of their locations. Figure 2 is a SolidWorks model of the design of the artifact

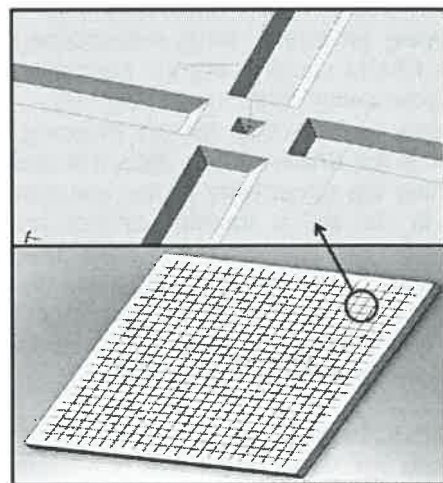


FIGURE 2. SolidWorks model of the Artifact

Three sapphire spheres will be attached to the bottom of the artifact for the kinematic coupling system.

A separate stage will be designed that will include three sets of v-grooves allowing for the rotation and displacement of the artifact. In order to accurately displace the artifact by 1mm, a pin system will be used as shown in Figure 3.

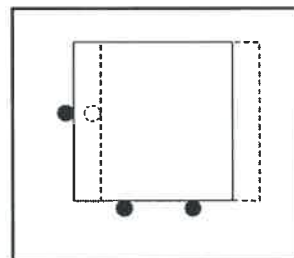


FIGURE 3. Three point pin system to displace the artifact by 1mm

When the wafer is placed onto the stage, it will be aligned against the pins.

CURRENT WORK

The preferential etching process must be thoroughly explored to achieve smooth etched surfaces and a uniform pattern.

Two well known silicon etches are Potassium Hydroxide (KOH) and Tetramethyl ammonium hydroxide (TMAH). It was decided to use TMAH because KOH forms many hydrogen bubbles which can cause surface roughening. This may be accounted for by stirring the etchant bath; however, too much stirring may cause non-uniform etching. The temperature of the etch bath and stirring speed also effects the surface finish. Generally, Isopropyl alcohol is mixed into the etch bath to make the etchant less aggressive, thus etches smoother [7].

An n-type [100] oriented silicon wafer was patterned with a UV stepper and a SiO₂ hard mask was formed. The wafer with 200nm square features was etched to obtain data on this process. TMAH (25% concentrated) was used for the etchant bath and was heated to 90°C using a hotplate with a temperature probe. A magnetic stirrer was immersed in the etchant and the hotplate stirring speed was set to its lowest value, 50 RPMs. The etchant was covered with a glass plate to capture the evaporated solution. An AFM image and surface analysis was acquired of the sidewall of an etched pit. Figure 4 shows the AFM image of the pit and the analysis of the sidewall.

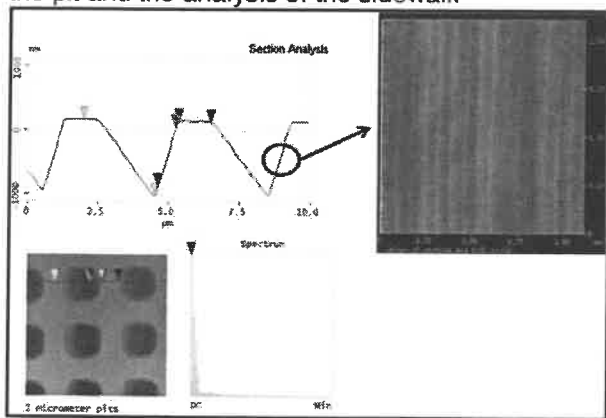


FIGURE 4. AFM image of etched pit (TMAH 25% at 90°C)

Notice in the above figure that the left side of the pit includes imaging of the tip, however, the left side is a sufficient image. In the future, a higher aspect ratio tip will be used to eliminate this effect. The RMS roughness value of the sidewall was approximately 1.099nm. This

value is at the edge of the resolution of the AFM. Better imaging techniques will be employed later. It does show the high quality of the surface finish. Different concentrations of IPA and etchant temperature will be studied to achieve the best surface finish.

The hard mask that will be used is thermally grown SiO₂. The advantages of using this hard mask are it protects the back side of the wafer, etches extremely slow in TMAH, and forms a non-porous film. The depth of the truncated pyramids is planned to be 2828 nm.

The features of the mask must be aligned with the flat on the wafer. If it is misaligned by 1°, it will etch into other crystal planes of the lattice and cause roughening of the surface [7]. It is important to have a patterning technique that can resolve 4000nm and pattern uniformly. A UV stepper at UNC Charlotte will be used. To assure proper positioning of the mask, alignment marks will be included on the mask.

Dry etching of the hard mask is ideal because of the vertical sidewalls it creates; however, the hard mask can't be etched all the way using dry etching. If the silicon is exposed during the etching, it could cause damage to the surface from ion bombardment. The mask will be etched mostly by dry etching and the remaining layer will be etched using a buffered oxide (BOE) wet etch. BOE does not attack silicon and will not damage the surface.

FUTURE WORK

The design of the kinematic coupling system incorporated with the three pins to assure repeatable positioning will have to be designed very carefully.

The 101.6mm substrate planned to be used to etch the pits will have to be cut into 25.4mm squares using a diamond saw. It is desired to have the edges aligned with the patterns and that are square with each other. The wafer will have to be rotated to be cut in both directions. The squareness of the cuts will be studied and optimized.

It is known that when silicon is exposed to air it reacts with the oxygen present and forms an oxide layer referred to as "native oxide." Since the artifact will be employed as a transfer standard for nano-positioning instruments, the oxide layer needs to be stable. A small oxide

layer needs to be grown on the substrate so that the silicon will no longer react with the oxygen in ambient air. This will be observed in further depth.

The cleanliness of the artifact before it is placed on the SAMM stage is important. The artifact must be cleaned with a solution called "piranha," to eliminate any organic compounds present. Since there are no chemical hoods in the room with the stage, the artifact will have to be cleaned in the cleanroom and carried to the stage without contamination. A traveling nitrogen box will be used and decided upon. Since the artifact must be cleaned after the sapphire spheres are attached, the glue to the used must be resistant to the "piranha." Phosphosilicate glass (PSG) will be used to glue the sapphire spheres onto the bottom surface of the artifact.

Hopefully this artifact will be measured using other systems. To assure no damage to the wafer, a traveling box will be designed and constructed that is able to be taken on a plane.

A procedure must be developed to find the cavity of the pits and correct for the XY mapping errors of the SAMM stage. The pits will be scanned using the AFM and the data will be recorded into a software program that will do a least square fit on all four planes and find the intersection point will be found. This algorithm will have to be developed.

N-type wafers with low doping concentration will be used to practice the etching process and to develop controllable etch rates. After optimal parameters are observed, the process will be used to fabricate the artifact. The corners of the pyramid may be rounded due to undercutting. Additive features on the mask may need to be considered and included to correct for these rounded etches.

CONCLUSION

It is desired to build this macroscopic transfer standard to correct for the systematic errors in SAMM and hopefully measure the artifact with other nano-positioning machines and compare results.

All of the variables in the etching process will be examined in great depth first. It is possible that a second mask will have to be made to account for the undercutting that may cause rounding of

the edges. The imaging process will be optimized as well. After sufficient pits are made, an algorithm to find the cavity bottom will be developed and initially modeled using MatLab.

After the etching process is completely developed, the other issues will be studied. This design is preliminary and is subject to change.

ACKNOWLEDGEMENTS

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PROCESS MODELING OF DIP PEN NANOLITHOGRAPHY LINE WRITING FOR ACCURATE LINE WIDTH CONTROL IN NANOMANUFACTURING

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INTRODUCTION

In this paper we describe process modeling of Dip Pen Nanolithography (DPN) line writing that enables control of line width within a few molecular dimensions. This capability has a number of potential applications such as writing a specified number of bio-molecules on a target substrate and fabricating several types of nanoscale features [1]. The realization of mass manufacturing in these applications is hindered by a lack of theory and practice that prevents bridging the gap between research and industrial-scale manufacturing. The gap requires precision manufacturing process control, and precision equipment/tooling to enable this transition. We first address process modeling to enable subsequent definition of equipment/tooling performance specifications.

Process modeling is an important element of the transition from nanofabrication to nanomanufacturing. This modeling is required for process performance prediction and control. Apart from prediction, models are also helpful in identifying the critical input parameters and the sensitivity of the process to those parameters. This further aids identification of areas wherein resources/effort would have the most impact upon process optimization. Current knowledge and practice in DPN line writing is however limited to an extent that accurate process control is difficult. The difficulty arises from the following technology/knowledge gaps:

1. Inability to predict performance without performing initial semi-empirical model fitting
2. Inability to model the meniscus as a finite ink reservoir instead of an ideal ink source
3. Not accounting for all the steps involved in the ink transport mechanism
4. Lack of knowledge about rate limiting steps and the corresponding process variable limits (e.g. maximum tip speed)
5. Lack of knowledge about relative sensitivity of process parameters

In this work, a model for DPN line writing has been developed to overcome these deficiencies.

SYNOPSIS OF THE DPN PROCESS

Dip Pen Nanolithography is a patterning process which may be used to write nanoscale features with a feature size resolution of 15 nm. In DPN, the "ink" material is written upon a substrate through the interactions of a sharp AFM tip, a meniscus and the substrate. Fig.1 shows the schematic of a DPN line writing process. An inked tip is dragged along the substrate surface to write lines. Transport of ink from the tip to the substrate is facilitated by the formation of a water meniscus at the tip-substrate contact. The meniscus forms as a result of capillary condensation [2]. The water meniscus acts as the "effective" medium for the transport of ink from the tip to the substrate. The ink transport mechanism may be separated into three distinct steps: (i) ink dissolution from tip into the meniscus, (ii) diffusion through the meniscus and (iii) diffusion of ink on the substrate surface; leading to formation of self assembled monolayers (SAMs) of ink molecules on the substrate. The size of the written feature depends on the characteristics of the reservoir (meniscus) and ink transport. A controlled quantity of ink may be delivered to the substrate by controlling the parameters which affect the meniscus geometry and the rate of the transport steps. Herein, we discuss the effect of tip speed and relative humidity on line width.

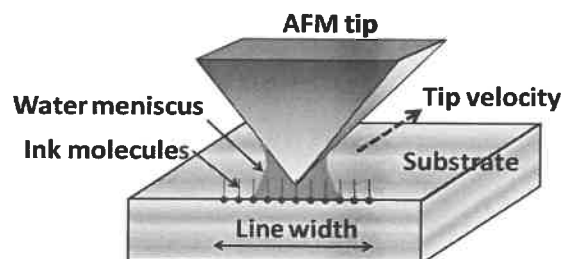


FIGURE 1: DPN line writing schematic

Mass Transport Steps

The three ink transport steps are illustrated in Fig. 2. The first step involves dissolution of ink from the ink-coated tip into the water meniscus at the tip-meniscus boundary. The next step involves diffusion of ink from the tip side to the substrate side along the meniscus under a concentration gradient. The final step involves formation of a SAM through diffusion of ink molecules on the solid substrate surface. Any of these steps may be the rate limiting step depending upon operating conditions. For example, for high tip speeds, meniscus diffusion may be the slowest step, thereby yielding "drying up" of ink in the meniscus. Our modeling may be used to identify which step is the rate limiting step for given process parameters.

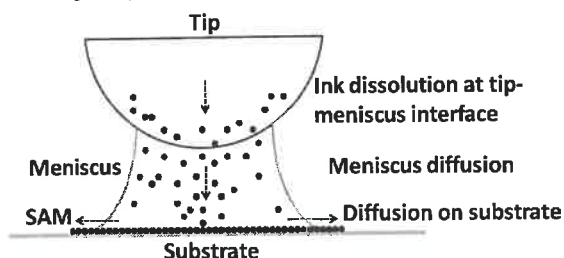


FIGURE 2: Schematic of mass transport steps

MODEL OF THE LINE WRITING PROCESS

Meniscus Model

The meniscus is formed via capillary condensation of water vapor at the tip-substrate contact. The curved liquid-vapor boundary yields a change in vapor pressure and causes condensation of water vapor per the relationship given in Kelvin's equation.

$$\ln \frac{p_v}{p_s} = \frac{\sigma V_m}{R_g R_c T} \quad (1)$$

Here, R_c is the radius of curvature of the meniscus, σ is the surface tension and V_m molar volume of water, R_g universal gas constant, T temperature and p_v/p_s the relative humidity.

The local meniscus curvature may be obtained as a function of ambient conditions using Eq. 1. To obtain the meniscus profile, the tip end has been modeled as a sphere. A toroid has been used as a "first pass" approximation of the meniscus geometry [2]. Meniscus properties such as the contact area, volume, contact radius and height may then be obtained from the profile. Fig. 3 shows how the contact radius and meniscus height increase with an increase in relative humidity.

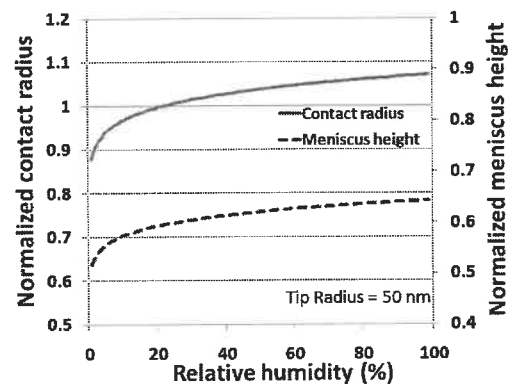


FIGURE 3: Effect of relative humidity on meniscus radius and height

Ink Transport Model

The meniscus has been further approximated as a cylinder in order to facilitate an elegant analytical model for the ink transport mechanism. The radius of the cylinder is obtained from the contact area and the height from the volume of the meniscus. Neglecting the initial transients, a steady state analysis of the line writing process has been completed. Ink diffusion through the meniscus has been modeled as 1-dimensional Fickian diffusion under a concentration gradient along the length of the cylinder. Ink dissolution at the tip-meniscus boundary may be incorporated as a boundary condition to this diffusion problem. At the other end of the meniscus, a concentration boundary condition is applied with an unknown concentration, C_o . Diffusion of ink on the substrate leading to SAM formation is solved by considering 1-D diffusion with a source concentration (as yet unknown, C_o) and a moving boundary condition. The diffusion model and boundary condition are given respectively in Eqs. 2 and 3.

$$\frac{\partial C}{\partial t} = D \frac{d^2 C}{dx^2} \quad (2)$$

$$-D \frac{\partial C}{\partial l} = k \frac{ds}{dt} \quad (3)$$

In these equations, C is the ink concentration, D is the ink diffusivity over SAM, k is a proportionality constant that depends upon ink molecule size, x is the direction perpendicular to tip velocity and s is the position of the SAM boundary on the substrate.

Solving Equations 2 and 3 yields the rate of SAM growth as a function of the unknown concentration C_o as shown in Eq. 4.

$$\frac{ds}{dt} = \lambda \sqrt{\frac{D}{t}} \quad (4)$$

Equation 4 is subject to the constraint that λ satisfies Eq. 5.

$$\sqrt{\pi} \lambda \text{erf}(\lambda) e^{\lambda^2} = C_0 / k \quad (5)$$

If the time interval for diffusion is known, the line width may be obtained by solving Eq. 4. To obtain the time interval, we considered that diffusion along a line perpendicular to the tip velocity takes place only as long as the meniscus is present over the corresponding "source point" as shown in Fig. 4.

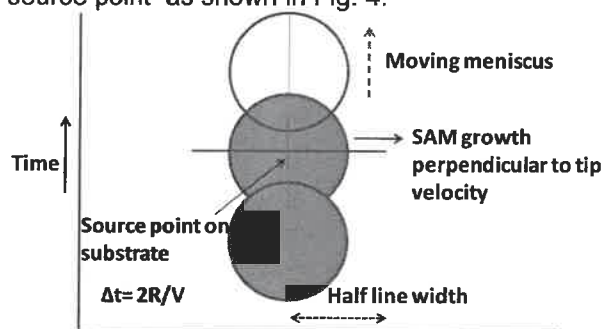


FIGURE 4: Model of meniscus motion and diffusion time

Under this condition, line width may be obtained as a function of the unknown concentration. Mass conservation is used to eliminate the unknown concentration and obtain the line width as shown in Eq. 6.

$$2vs = \dot{N} A \quad (6)$$

In Eq. 6, v is the tip velocity, $\dot{N}$ the ink molecular flux through the meniscus and A the area of each molecule. The rate limiting step can be identified by the ink concentrations at the meniscus boundaries. Ink dissolution or meniscus diffusion is the slowest step depending on whether the concentration at tip end or the substrate end is non-positive, respectively. If both the concentrations are positive, diffusion on substrate is the slowest step.

Model Predictions

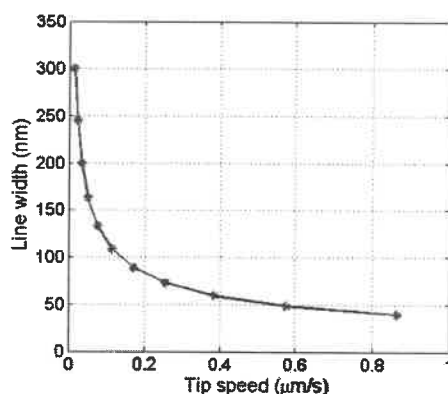
This model has been used to predict the line width as a function of tip speed and relative humidity. MHA ink (16-Mercaptohexadecanoic acid) properties have been used in simulations run for operating temperature of 298 °K. Meniscus concentrations were found to be

positive indicating that diffusion on substrate is the slowest step. The model predictions in Fig. 5 show that changes in tip speed have more effect than changes in relative humidity. The line width may be changed from approximately:

(a) 300nm to 50nm by changing the tip speed from 0.1 $\mu\text{m/s}$ to 1 $\mu\text{m/s}$

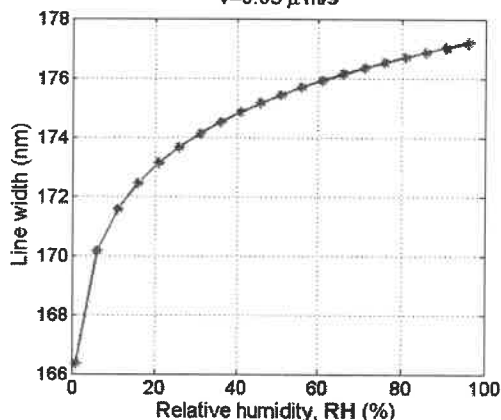
(b) 6 nm by changing relative humidity from 15% to 95%

These results indicated that coarse control of line width may be performed by controlling the tip speed and fine control may be obtained via control of relative humidity.



(a)

$v = 0.05 \mu\text{m/s}$



(b)

FIGURE 5: Predicted effect of (a) tip speed and (b) relative humidity on line width

EXPERIMENTS

Experiments were performed on a gold substrate with MHA ink in a Nanolnk DPN 5000 system [3]. The system is shown in Fig. 6. Silicon nitride AFM tips of type 'A' from Nanolnk were used. They possessed a nominal cantilever length of 200 μm , width of 45 μm and spring constant of 0.041 N/m [3]. Inking was done by dipping the tip in 5 mM solution of MHA

ink in acetonitrile for 15 seconds and then drying in a stream of nitrogen gas. Lines were written at different tip speeds and relative humidity. The Nanolnk InkCAD software was used to control the tip speeds and relative humidity with an accuracy of 1%. An AFM scan of the written lines is shown in Fig. 7. The lines widths, measured using the Nanolnk line-cursor measurement tool in InkCAD are shown in Fig. 8.

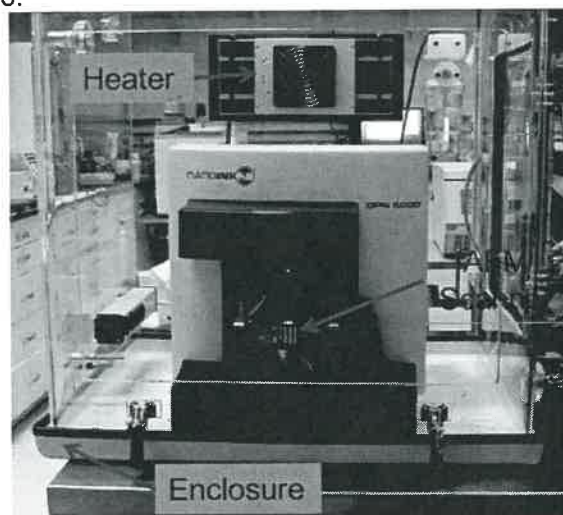


FIGURE 6: Nanolnk DPN 5000 system used for the experiments

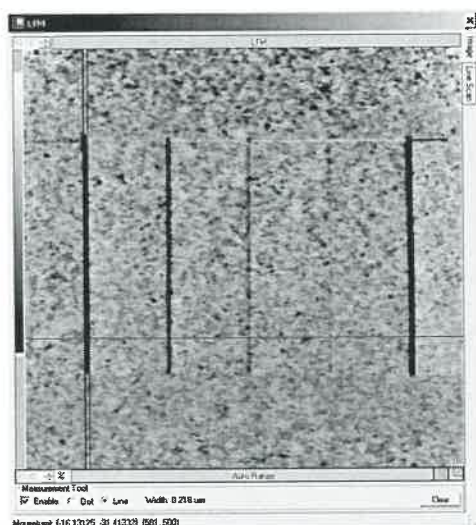


FIGURE 7: AFM scan of written lines

Experimental values for tip speed are within 5% of the model predictions. This trend we observe indicates that the product of tip speed and width is constant. Considering Eq. 6, this implies a constant ink flux and therefore a steady-state

assumption for the process is justified. The effect of relative humidity is found to be much higher than predictions especially at high humidity suggesting a larger meniscus which is not restricted to the spherical probe tip, but rather reaches beyond this area.

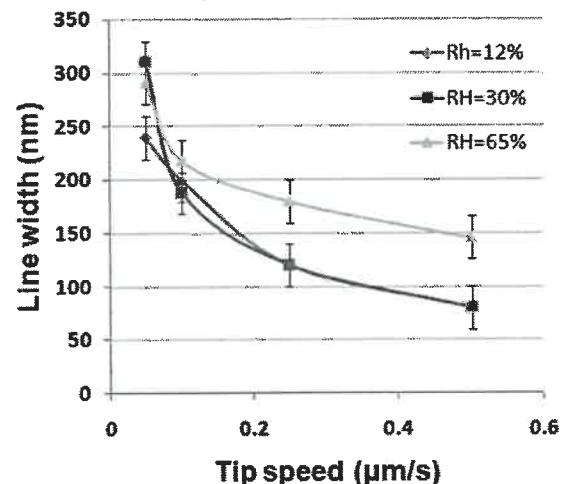


FIGURE 8. Effect of tip speed and relative humidity on line width

CONCLUSIONS

We have shown early work that demonstrates how one may model a DPN line writing process and thoughts on how to conduct precision process control. We identify diffusion on the substrate as the slowest step in the process. We have also identified the means of coarse line width control via control of tip speed and fine width control via control of relative humidity.

Future work will encompass a more sophisticated parametric model of the process that includes temperature and tip geometry. This knowledge base would then be used to develop error and yield models for multiple tips, thereby enabling some measure of process control in massively parallel DPN process.

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EVALUATION OF A PROTOTYPE CMP MACHINE FOR IMPROVING THE GLOBAL PLANARIZATION BY MEMS

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ABSTRACT

Planarization using Chemical-mechanical Polishing (CMP) is a key technology for fabricating advanced Integrated Circuits (IC) and Micro Electro-mechanical Systems (MEMS). In MEMS fabrication, CMP is used for various applications with an emphasis on surface micromachining. However, due to the much larger structures, wafer level global planarity requirements are harder to fulfill. There are two effects competing against each other. On one hand, creating the process pressure by exerting a force to the wafer at its center favors a greater inside pressure. On the other hand, due to the CMP pad's compliance, the pressure is distributed asymmetrically, resulting in a maximal pressure peak at the wafer's rim. To achieve a constant removal rate and thus an optimal global planarization, the pressure should be uniform over the entire wafer. However, this may only be achieved if the wafer profile is adjusted appropriately. One way to adjust the wafer profile and, if necessary, even the profile of the CMP plate is to use appropriate tools. They are available in a prototype CMP machine specifically developed for this task. This paper presents an evaluation of the capabilities for such a machine prototype regarding the global planarization.

INTRODUCTION

CMP, originally developed for the semiconductor industry, is widely used in the fabrication of MEMS. However, MEMS often feature larger geometries compared to integrated circuits, in order to fulfill mechanical functions [1, 2]. Particularly for fabricating MEMS with a great building height requiring High Aspect Ratio Micro Structure Technology (HARMST), there are two challenges when applying CMP [3]. The first is achieving a local planarization, i.e. a

minimal step height between structures and embedding material. The second is reaching a global planarization, which means that the material removal during the CMP process between the wafer's center and its rim is the same (Fig. 1).

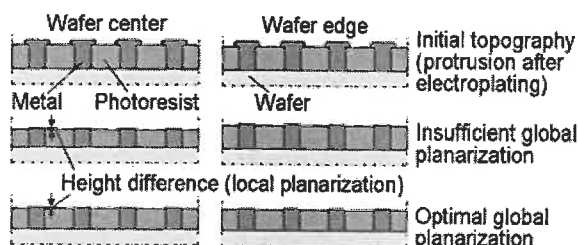


FIGURE 1. Global and local planarization by CMP of MEMS

In general, a CMP process is carried out on a rotary table. During the process, a range of variables are affecting each other substantially [4]. The main factors are the wafer holder, the polishing pad, and the process parameters (Fig. 2).

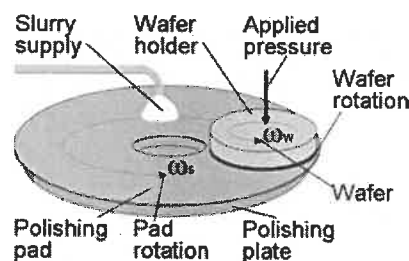


FIGURE 2. CMP process [4]

While an optimal local planarization typically is accomplished through appropriately matching the CMP slurry, the pad features, and the applied process pressure, reaching a global planarization much more strongly depends on the machine itself and its mechanical properties.

To achieve a constant removal rate and thus an optimal global planarization, the pressure must be uniform over the entire wafer. One way to optimize the global planarization is to use appropriate tools. They are available in a CMP test machine specifically developed for this task. The machine features a significant level of flexibility in order to provide an optimal global planarization as well as a sufficient surface quality for a wide spectrum of MEMS applications considering the materials involved and the magnitude of the structure geometry.

MACHINE CONCEPT

To achieve a high uniformity of the pressure over the entire wafer, a two-prong approach was pursued: to adjust the contour both of the wafer holder, as well as of the plate. To adjust the wafer contour, a special piezo wafer holder developed by Noah Technology (former IGAM) in Magdeburg, Germany was used. Figure 3 schematically depicts its adjustment capabilities.

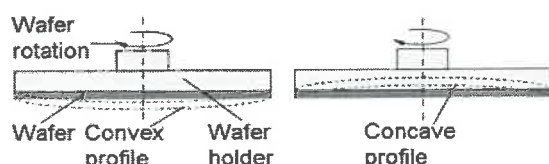


FIGURE 3. Deformations of the wafer holder

To perform the actuation, the wafer holder consists of concentric piezo elements, with a piezo disk at the center, surrounded by concentric piezo rings (Fig. 4).

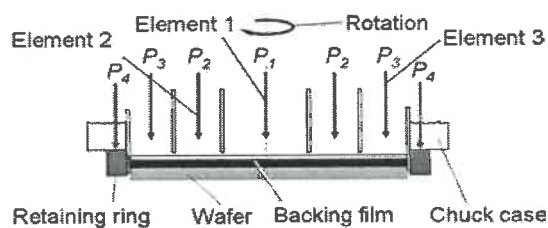


FIGURE 4. Schematics of piezo chuck

By appropriately exciting the piezo elements, either a convex or a concave wafer contour can be achieved. The total topography variation of the piezo chuck is specified to be $\pm 8 \mu\text{m}$. In addition, a retaining ring is equipped with screw adjustable spring bolts. As a result, two or three independent pressure elements, respectively, and an adjustable retaining ring may be adjusted appropriately. Holding the wafer during the CMP process is accomplished either through a backing film or through vacuum suction.

Not only the wafer chuck, but also the polishing plate may change its contour (Fig. 5). As in the case of the wafer holder, a flat plate, as well as various degrees of concave or convex contours may be chosen. The adjustment is accomplished pneumatically. Two plates with two different ground states were supplied: one is flat and only capable of changing into convex contours. The other one is concave, allowing a concave (Fig. 5a), flat (Fig. 5b), or convex a contour ranging from convex to concave (Fig. 5c). The specification for the plate with a concave contour in its unpressurized state is as follows: at the plate center, the maximal concave profile results in a features a protrusion of $50 \mu\text{m}$.

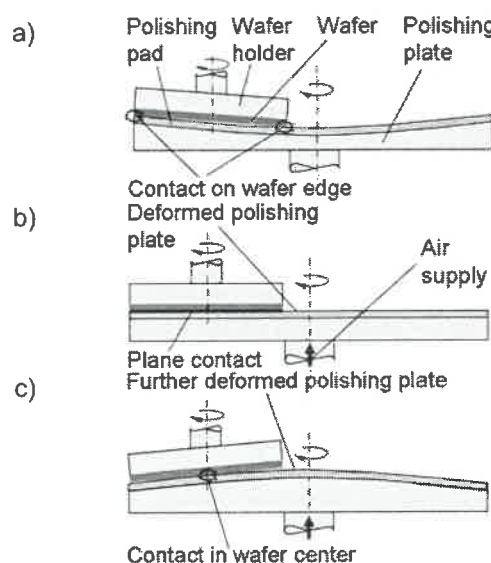


FIGURE 5. Deformation of polishing plate: a) initially convex, b) deformed plane and b) deformed concave state

The adjustable, pneumatically actuated plate as well as the whole CMP machine were developed by FLP Microfinishing, Zörbig, Germany. The Noah piezo wafer holder is integrated in it. Figure 6 depicts the machine at its test location inside the cleanroom of the Institute for Microtechnology (imt).



FIGURE 6. CMP machine in the imt's clean room

MACHINE VERIFICATION

As a preparation for evaluating the machine capabilities with respect to global planarization, measurements of real tool deformations were conducted. First, the warping of piezo elements on the wafer holder was measured (Fig. 7).

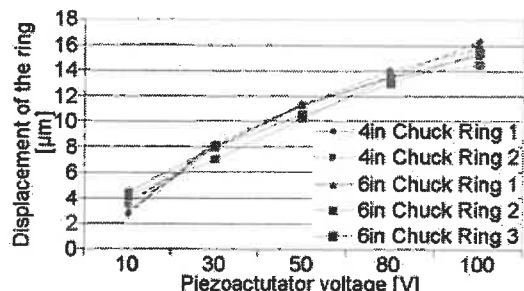


FIGURE 7. Displacement of the piezo elements

The results showed an achievable maximal disk / ring expansion of 16 μm by all rings and both chucks (4 in and 6 in). Thus, the specification for a chuck adjustment of $\pm 8 \mu\text{m}$ was fulfilled. Although all elements should elongate equally, a minor deviation in elongation between the individual elements was measured and averaged $\pm 1 \mu\text{m}$.

Polishing plate measurements were conducted on a plate which was flat when unpressurized. For the tests, two CMP pads were used: a soft felt polishing pad and a hard polyurethane CMP pad (Fig. 8). The maximal achievable displacement of the plate was greater than 100 μm . Since the plate actuation range is independent of the original state of the plate, the results show that the specified limits (+50 μm to -20 μm) can be achieved.

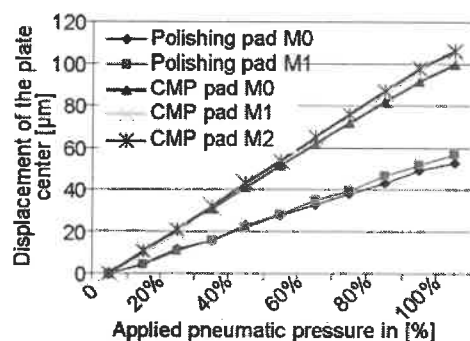


FIGURE 8. Displacement of the polishing plate

EXPERIMENTAL

CMP tests were carried out to investigate the removal. Each tool set (wafer holder and

polishing plate) was activated separately; this way the respective influence could be tested. For the tests, Si wafers coated with 50 nm Cr and 400 nm NiFe were used. The first tests were done with a piezo chuck and a solely excited center disk (Fig. 9a). The excitation of 100 V is accordant to an element elongation (r_1) of 16 μm . This led to an increasing of the applied pressure in the center of the wafer and a corresponding concentration of the removal (Fig. 9b). To expand the removal from center to the wafer rim after four minutes of processing, a second element was excited additionally (r_1, r_2). This generated a removal broadening from the wafer center (after ten minutes). In order to remove a remaining coating on the wafer edge, element 3 is excited solely. After 20 minutes the whole wafer coating was removed.

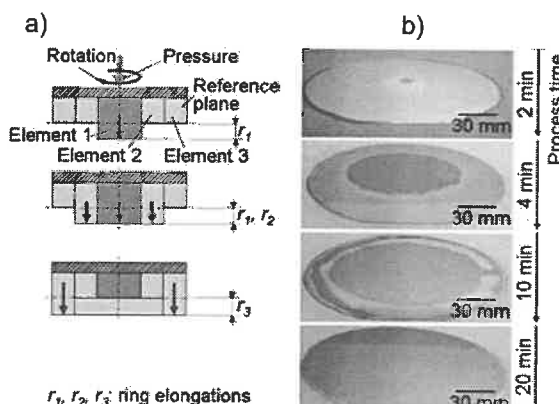


FIGURE 9. Effect of the wafer holder: a) chuck engagement and b) material removal as a function of machining time

By engaging the piezo-actuated disk at the center or the edge of the wafer holder, a controlled removal was achieved. Both chucks (4 in and 6 in) were subjected to the same investigations, showing correspondent impact on the material removal.

For the tests of the polishing pad, a plate initially plane was pneumatically warped to the maximal curvature. As a result, a removal localized at the wafer's center was achieved (Fig. 10).



FIGURE 10. Effect of polishing pad

After the tests with the separately actuated tools, tests with simultaneously exciting both, the wafer holder and the polishing plate were conducted. This led to a noticeable decrease in

polishing time and thus 30 percent less process time. Table 1 depicts the results. The wafer surface after CMP was compared to the state as delivered, which was polished for Si and lapped for Al_2O_3 .

TABLE 1. Results with coated wafers

Wafer	Si		Al_2O_3	
	Polished	CMP	Lapped	CMP
Average thickness [μm]	527.6	517.4	516.1	511.1
Max. difference in thickness [μm]	8.08	6.25	14	10
Average thickness deviation [μm]	8.1	1.5	6.8	4.4

After the functionality and capability tests with bulk wafers, first preliminary processes on patterned wafer were conducted. For this purpose, the electroplated permalloy structures embedded in photoresist (AZ) on the Al_2O_3 wafers were used. The structure height distribution on the wafer was measured after electroplating. With a tool adjustment corresponding to these measurements, the CMP process was carried out. The main challenges were to planarize both larger and smaller structures without damage and delamination, and still achieving a global planarization. Planarization results achieved with patterned wafers are shown in Table 2.

TABLE 2. Results with patterned wafers

Wafer	Si	
	Before CMP	After CMP
Average structure height [μm]	>30	18 - 23
Max. height difference [μm]	10 - 20	8 - 9
Average step height [μm]	4 - 8	2.0 - 2.5

Fig. 11 shows a structure height distribution over the wafer before and after CMP. After electroplating, the structures on the wafer edge were much higher than in the center (Fig. 11a). After CMP, an improvement in global planarization is clearly visible with an average step height between the structures of 2.5 μm (Fig. 11b).

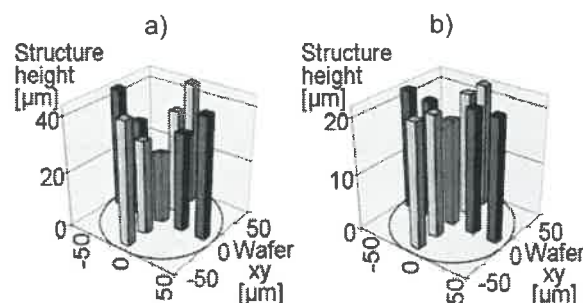


FIGURE 11. Global planarization result on a patterned wafer (xy-plane): a) before CMP and b) after CMP

CONCLUSIONS

The tests showed that differences in material removal between the wafer center and the wafer rim could be achieved by appropriately adjusting piezo-actuated elements and deforming the polishing plate. Improvement in global planarization could be determined by both coated and patterned wafers. Functional tests showed a high application potential of both tools, piezochuck and polishing plate, for improving global planarity through a CMP process.

ACKNOWLEDGEMENTS

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IN PROCESS GRINDING WHEEL SURFACE INSPECTION USING MACHINE VISION

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INTRODUCTION

The purpose of this work is to study the changes in wheel topography in cylindrical grinding. This seemingly modest goal is complicated by the overwhelming number of parameters and physical influences on the material removal process of abrasive grinding. To date, no one has successfully captured the effects of all parameters or even a small subset of the relevant input parameters such that an arbitrary grinding process may be modeled and the wheel wear accurately predicted. In this work cylindrical grinding operations are investigated on an ultra-precision grinding machine that features a second-generation instrumented spindle allowing real-time, high-bandwidth force feedback from the cylindrical grinding process and an integrated camera for wheel inspection.

For the early stages of this work introduced here, only part of the instrumentation will be discussed. This part is the integrated camera for in-process wheel inspection.

PROJECT MOTIVATION

Compared to other cutting processes the grinding process is influenced by a much larger number of parameters. This has lead to enormous effort being poured into trying to understand the material removal mechanisms of the interesting, but complex, grinding process. Figure 1 shows a schematic of the grinding process and illustrates some of the main grinding inputs, interactions, and outputs. For this work the grinding wheel condition is analysis through the use of a vision system. Adding complication are the many nonlinear relationships between the grinding process inputs and outputs. These nonlinearities are even further convoluted by time-varying conditions such as wheel wear, chatter, and disturbances. Today, selecting proper grinding parameters and compensating for the numerous nonlinearities remains a perplexing problem. Due to these arguments many believe the grinding pro-

cess to be an art requiring skilled operators more than a science to be explained.

Other motivating factors for this work include unique equipment at our disposal, and not available at any other research university shown in Figure 2. This includes a second generation instrumented grinding spindle, our lab's recently acquired Moore Nanotechnology Systems UPL 450 and an integrated camera. Essential parameters for the completion of the traverse grinding model can be experimentally varied under the most precise control using this new hardware.

EXPERIMENTAL SETUP

Grinding processes in general are quite complex with many parameters that are usually hard to quantify and essentially stochastic in nature. As a result, successful workpiece quality (e.g., surface finishes and subsurface damage) have a heavy reliance on skilled operators. In spite of the complexities these processes are indispensable for achieving precise tolerances when applied as a finishing step. In advanced engineering materials including ceramics and hardened steels, the grinding process is the only option for shaping the workpieces into final products. Despite this, grinding remains one of the most difficult and least-understood machining processes. It also accounts for the major portion of the machining cost [1].

One of the reasons the grinding process is so convoluted is the surface of the grinding wheel contains many cutting points in the form of abrasive grains held together in a softer matrix. These grains are made of hard and tough ceramics or crystalline materials that are used to penetrate and cut softer materials. Therefore, the grinding wheel can be thought of as a multi-point cutting tool. Unlike the easily-characterized single point cutting tools, abrasive grains are geometrically undefined in both location and shape. This makes it difficult to know the true conditions of

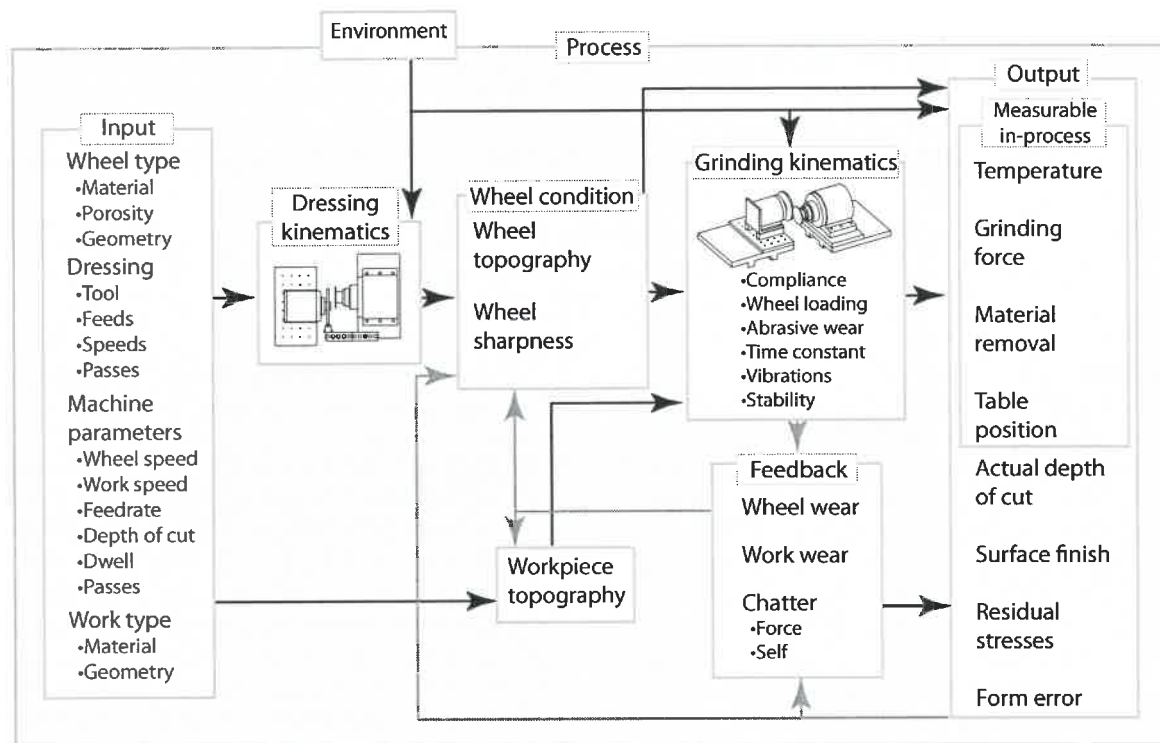


FIGURE 1. Schematic of the grinding process.

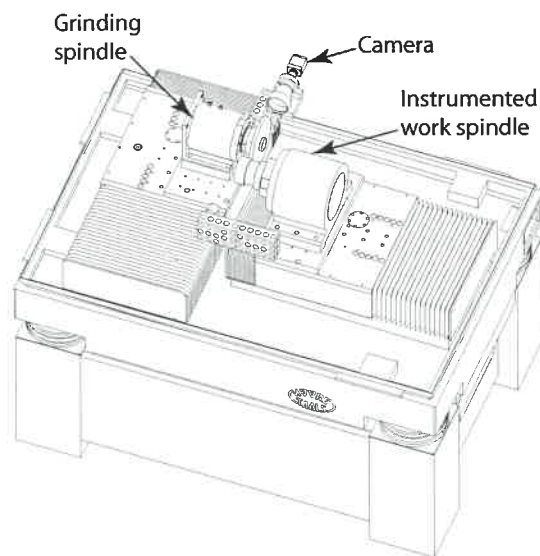


FIGURE 2. Ultra precision lathe UPL 350 test bed mounted with an instrumented work spindle, grinding wheel spindle synced with a camera.

the grinding wheel surface with any degree of certainty.

For decades it has been known that the dressing variables have a big impact on the grinding wheel condition and grinding process efficiency [2]. However, the actual condition of the wheel surface, in particular the cutting ability, is an undefined grinding parameter, and the question becomes how may it be defined as a number? When we can place a reliable, repeatable and reproducible, value on the condition of the wheel then the effects caused by this changing condition number can be evaluated [3]. Different techniques have been used to determine the conditions and abrasive grain count of the grinding wheel [4, 5, 6].

PROCEDURE AND PRELIMINARY RESULTS

A subsection for this work involves a vision system which is utilized for in process wheel surface inspection. The camera is timed to the grinding spindle using triggering off the high edge counts of the spindle's encoder. This allows for repeatable images on the same location on the surface of the wheel. Previous images are stored on the computer hard drive and can be used to build a

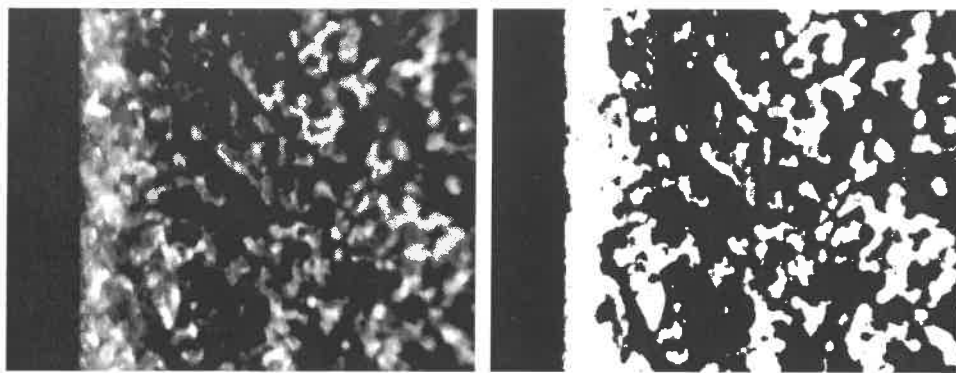


FIGURE 3. Section of the aluminum oxide wheel captured, 7.6mm by 5.7mm, triggering the machine vision camera (left). Binary image processed through MATLAB showing the cutting grains on the surface of the aluminum oxide wheel (right).

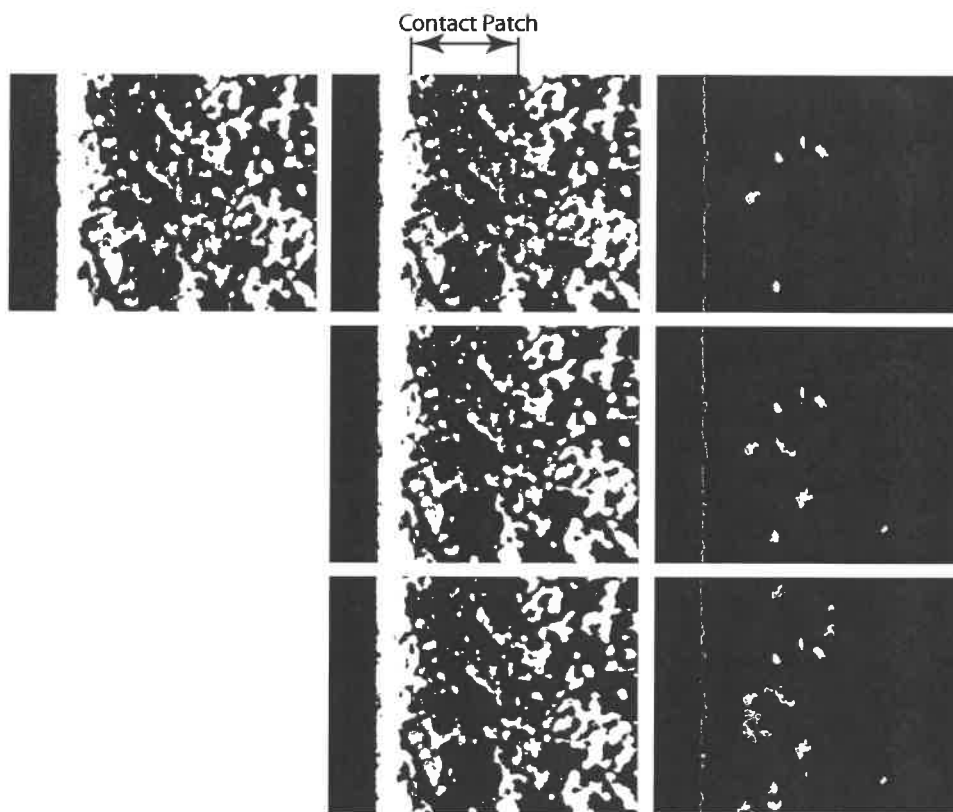


FIGURE 4. Wear of aluminum oxide wheel. The upper left corner in the first column, first row shows the binary section of the grinding wheel just after dressing. The second column, first row show the wheel after eighteen passes, second row is thirty-six passes, and the third row is fifty-four passes. The third column is the first column subtracted by the second column. For this group of tests only a small section of the wheel made contact with the work, plunge grind.

library of images for training that particular wheel.

At full frame an image is composed of 307,200 pixels, each pixel having an intensity value from 0 to 255 or two to the eighth possible values. The data can be arranged in a matrix form where the pixels form the rows and incoming images form the columns. In this form the previous data can be quickly compared to new incoming data. Decreases in pixel intensity can be shown to result in grain breakage or removal while increases of intensity can be shown to result in embedded metal particles. Using previous grinding cycle data with the same wheel can result in a learned optimal time for dressing that particular wheel

The results introduce in this work used a soft aluminum oxide wheel, 38A80-F12VBEP, with a mild steel to avoid loading and look only at the wheel wear over time of the process. The work is spun at an angular velocity of 300 RPM while the grinding wheel at 1800 RPM. A depth of cut of about 5 μm is used with a very fast feed rate to encourage wheel wear. For these initial tests a plunge grind is used with a thin workpiece so that the entire contact patch on the wheel of the wheel/workpiece is within the view of the camera.

A Prosilica GE680 camera with a 0.31 telecentric lens is used with a LED ring light between the camera and wheel. Using this setup and the camera software controls, a 7.6 mm by 5.7 mm section of the wheel is captured. The camera uses a gigabit ethernet interface to transfer the data to a frame grabber located inside a work computer. The data can then be processed and stored for wheel condition evaluations.

The grinding wheel is step into its depth of cut on the workpiece and then retracted with no spark-out. This is repeated twice more before the camera is triggered to capture an image. On the third pass the work is moved away from the wheel and the grinding wheel's angular speed is decreased to avoid blurring with the current camera's settings and lighting. The camera then captures an image like the one shown on the left in Figure 3. The collected images are then processed using the MATLAB toolboxes. Multiple images of the same section of the wheel allows for the condition of the surface of the wheel to be compared throughout the grinding process to examine wheel wear. Figure 4 shows one of the sections on the wheel which was triggered multiple times over the grinding process. The initial im-

age of the wheel is captured after dressing and used to make a comparison of the wheel condition throughout the process.

CONCLUSION

This work demonstrates a section of our lab's specialization in instrumentation and that vision data can be used as a tool for evaluating the conditions of the grinding wheel's surface.

However, these results only look at the surface condition for the wheel and do not tell us anything about the form of the wheel. If the wheel is lobed and the section of the wheel triggered for capture is in the valley of the wheel lobe, it may have a different impact on the workpiece. Another piece of the puzzle for this work evolves an indicator with a special carbide tip to resist wear and measure the form error of the wheel. Fusing the information between the two sensors will help establish a number for cutting efficiently of the wheel and the effects caused by this changing condition.

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MECHANOCHEMICAL SUPERFINISHING OF OPTICAL GLASS – SCRATCHLESS SURFACE FINISHING PROCESS –

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ABSTRACT

This paper describes a newly developed mechanochemical superabrasive stone containing CeO_2 abrasive. The effect of CeO_2 abrasive on their performances was evaluated through the superfinishing of optical glass. It was found that soft CeO_2 abrasive rubs the surface of optical glass producing smooth surfaces with fewer scratches than diamond abrasive. The addition of CeO_2 abrasive improves the surface integrity of optical glass because it reduces production of scratches by diamond abrasive.

INTRODUCTION

Abrasion machining using bonded abrasives provides rapid material removal and high dimensional accuracy. In addition, using mechanochemical stones [1], which have interfacial reaction with works, they provide better surfaces. On the basis of this way of thinking, some mechanochemical stones, containing cerium oxide, CeO_2 , have been developed [2, 3].

Authors have developed new vitrified-bonded superabrasive stones, which contains the abrasive CeO_2 , and have investigated the chemical reaction between CeO_2 and Fe theoretically and experimentally [4]. The abrasive CeO_2 is softer than the optical glass but reacts chemically with them [5]. Using these stones, it was possible to obtain a smooth surface with fewer scratches, which was not possible when using conventional diamond stones.

In this paper, the performances of CeO_2 stone, diamond stone, and diamond stone containing CeO_2 abrasive are evaluated and compared through superfinishing experiments of optical glass.

PERFORMANCE OF MECHANOCHEMICAL SUPERABRASIVE STONE

Test Stones

A Mechanochemical superabrasive stone was developed for superfinishing of optical glass. It consists of CeO_2 20000-grit size with vitrified bond. In this paper, this is referred to as CeO_2 stone. In addition, a mechanochemical superabrasive stone, which consists of CeO_2 of 20000-grit size and synthetic diamond of 4000-grit size with vitrified bond, was developed. This is referred to as SD/ CeO_2 stone in this paper. A conventional vitrified SD stone was also prepared to investigate the effect of the CeO_2 abrasive on the performance of stone.

Experimental Procedures

All tests were performed on a centerless flat surface lapping machine. The machine is shown in Fig. 1. The workpiece was face-finished by a rotating cup-shaped superabrasive stone with oscillating motion. Table 1 tabulates the superfinishing conditions that were used.

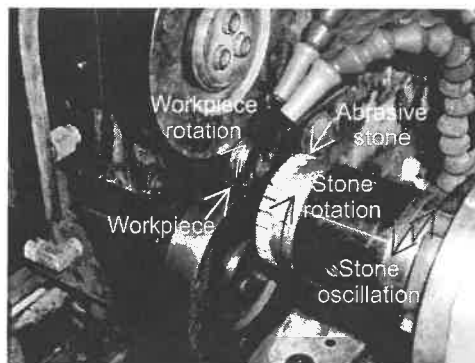


FIGURE 1. Centerless flat superfinishing using a straight cup-shaped abrasive stone.

Results and Discussions

CeO₂ Stone

If the CeO₂ abrasive actually reacts on an optical glass workpiece in the superfinishing process, the material removal rate of the stone containing the CeO₂ abrasive will obey the Arrhenius equation, because of the temperature dependence of chemical reaction rate.

$$k = A \exp\left(-\frac{E}{RT}\right) \quad (1)$$

where k is the reaction rate, A is the pre-exponential factor, E is the activation energy, R is the gas constant, and T is the absolute temperature.

The removal rate of the CeO₂ stone was analyzed based on this empirical law for coolant temperature from 283 K to 313 K. Fig. 2 shows the Arrhenius plot of the logarithm of the removal rate against the reciprocal of the

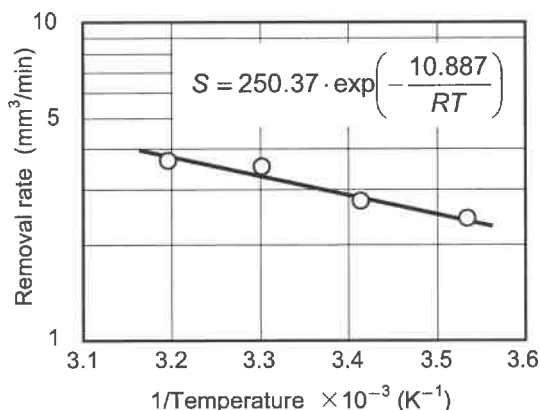


FIGURE 2. Dependence of removal rate on temperature of the superfinishing fluid.

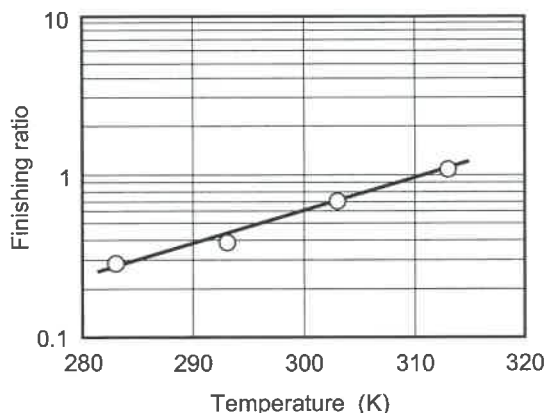


FIGURE 3. Dependence of finishing ratio on temperature of the superfinishing fluid.

TABLE 1. Superfinishing conditions.

Superfinishing pressure	0.17 MPa
Stone speed	42 m/min
Workpiece speed	79 m/min
Frequency of oscillation	6.7 Hz
Amplitude	0.5 mm
Superfinishing fluid	Dilute solution of rust inhibitor in water, Concentration=1%
Superfinishing fluid temperature	283-313 K
Superfinishing time	30 s
Workpiece material	BK7

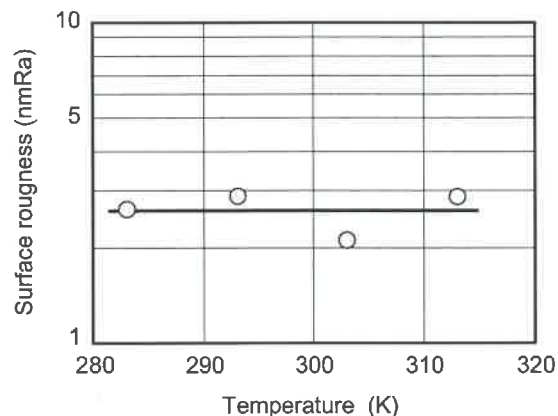


FIGURE 4. Dependence of roughness values on temperature of the superfinishing fluid.

temperature. The removal rate of the CeO₂ stone increase almost linearly as the temperature increases. Calculating the apparent activation energy from the slope of the straight line gives 10.9 kJ/mol for the CeO₂ stone.

Fig. 3 shows the finishing ratio against the temperature. The finishing ratio of the CeO₂ stone increase almost linearly as the temperature increases because of the increase of the removal volume and the decrease of the stone wear volume. These results indicate that the CeO₂ abrasive reacts on an optical glass workpiece in the superfinishing process.

Fig. 4 shows the roughness values, Ra, of the surfaces generated by the CeO₂ stone. The CeO₂ abrasive hardly scratches the surface of optical glass but only rubs it, because of its friability. Nevertheless, the CeO₂ stone effectively generates a smoother surface.

Fig. 5 shows the superfinished surface of BK7 with the CeO₂ stone. The surface has fewer scratches than conventional diamond stones. In some case, however, few scratches, which were generated with diamond abrasive during pre-finishing process, remains after superfinishing.

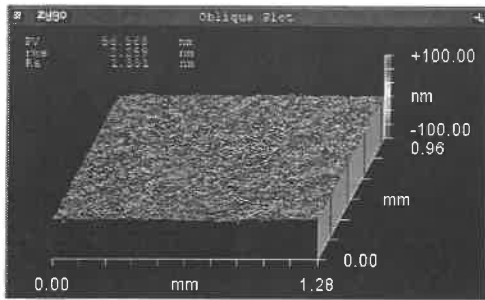


FIGURE 5. Superfinished surface of BK7 with CeO_2 stone.

$PV=56.6 \text{ nm}$, $rms=2.3 \text{ nm}$, and $Ra=1.5 \text{ nm}$.

SD/ CeO_2 Stone and SD Stone

As mentioned above, pre-finishing-process-oriented scratches sometimes remained after superfinishing with CeO_2 stone. To improve the integrity of the finished surface, it is necessary to reduce such scratches.

From this standpoint, we investigated the performances of SD/ CeO_2 stone and SD stone using method of experimental design.

Table 2 tabulates the superfinishing conditions that were used. Superfinishing conditions were arranged following orthogonal table of $L_{25}(5^4)$, i.e. four factors (superfinishing pressure P , stone speed V_2 , workpiece speed V_1 , and superfinishing time t) with five levels respectively. After superfinishing, removal rate S , wear ratio W_{ratio} , roughness value Ra , and scratch depth R_{p-v} were analyzed with multiple linear regression analyses.

The multiple-regression models obtained are as follows:

$$\text{removal rate } S \propto V_2^\alpha P^\beta t^{-\gamma} \quad (2)$$

$$\text{wear ratio } W_{ratio} \propto V_2^{-\chi} P^{-\delta} t^{-\varepsilon} \quad (3)$$

$$\text{roughness value } Ra \propto V_2^{-\phi} P^{-\eta} \quad (4)$$

$$\text{depth of scratch } R_{p-v} \propto V_2^{-\lambda} P^{-\sigma} \quad (5)$$

From these formulae, when superfinishing at the same superfinishing time, all of those finishing characteristics just depend on superfinishing pressure P and stone speed V_2 . Therefore, we can estimate optimum superfinishing condition to maximize removal rate with certain constraint conditions.

Fig. 6 shows an example of an optimum superfinishing condition of SD/ CeO_2 stone at $t=180 \text{ s}$, $Ra \leq 0.02 \mu\text{m}$, $R_{p-v} \leq 0.15 \mu\text{m}$, and $W_{ratio} \leq 0.5$. The optimum condition is $V_2=452 \text{ m/min}$ and $P=0.49 \text{ MPa}$, and the removal rate

TABLE 2. Superfinishing conditions for method of experimental design.

Superfinishing pressure	0.09-0.49 MPa
Stone speed	113-452 m/min
Workpiece speed	66-132 m/min
Frequency of oscillation	16.7 Hz
Amplitude	0.5 mm
Superfinishing fluid	Dilute solution of rust inhibitor in water, Concentration=1%
Superfinishing fluid temperature	293 K
Superfinishing time	60-180 s
Workpiece material	BK7

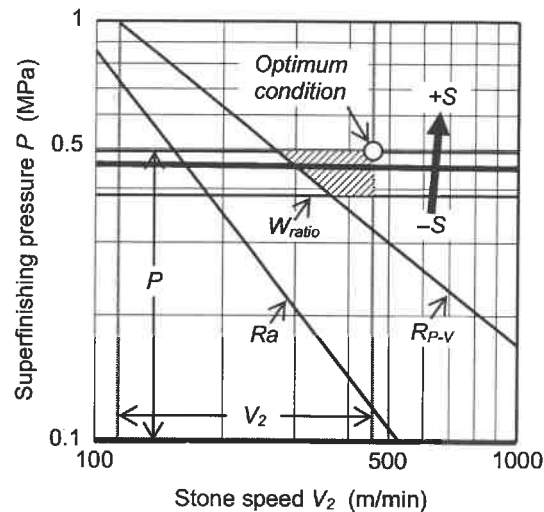


FIGURE 6. Optimum condition of SD/ CeO_2 stone.

$t=180 \text{ s}$, $Ra \leq 0.02 \mu\text{m}$, $R_{p-v} \leq 0.15 \mu\text{m}$, $W_{ratio} \leq 0.5$, $V_2=452 \text{ m/min}$, $P=0.49 \text{ MPa}$, $S=3.6 \text{ mm}^3/\text{min}$.

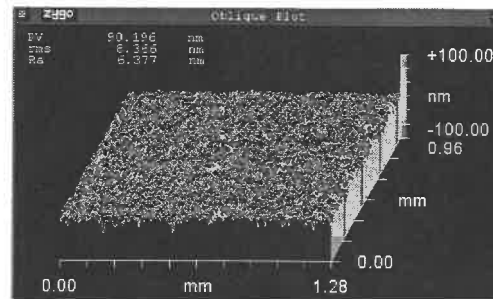


FIGURE 7. Superfinished surface of BK7 with optimum superfinishing condition of SD/ CeO_2 stone.

$PV=90.2 \text{ nm}$, $rms=8.4 \text{ nm}$, and $Ra=6.4 \text{ nm}$.

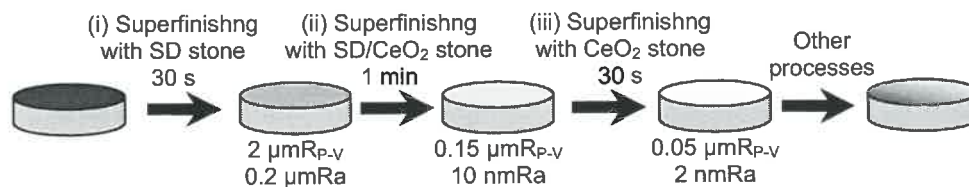


FIGURE 8. Scratchless surface finishing process with (i) SD stone, (ii) SD/CeO₂ stone, and (iii) CeO₂ stone.

estimated is $S=3.6 \text{ mm}^3/\text{min}$.

Fig. 7 shows an example of a superfinished surface of BK7 with optimum superfinishing condition of SD/CeO₂ stone. Using the optimum condition, it was possible to produce smooth surface without deep scratches in high removal rate and in one minute. This is probably due to both chemical removal with the CeO₂ abrasive and mechanical removal by the diamond abrasive.

SCRATCHLESS SURFACE FINISHING PROCESS

Fig. 8 shows scratchless surface finishing process with SD stone, SD/CeO₂ stone, and CeO₂ stone. This process does reduce process time and improves the surface integrity. Thus, this process maybe much useful to larger optical glass, such as reflecting mirrors of next generation extremely large telescopes.

CONCLUSIONS

The SD/CeO₂ stone produces smooth surface with less scratch, good roughness, and small stone wear, and high removal rate; that is, the SD/CeO₂ stone is suitable for pre-finishing.

The CeO₂ stone reacts with optical glass surface and produces a smooth surface with less scratch that is suitable for finishing process of reflecting mirrors.

Add to these results, the total finishing time of the process is shorter than two minutes.

Consequently, it is concluded that the process chain has a superior finishing performance.

ACKNOWLEDGEMENT

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FINISHING OF CaF_2 -GLASS WITH A NOVEL FINE GRINDING CONCEPT

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INTRODUCTION

The finishing of glass and ceramics is still comparatively time-consuming and expensive, which is a significant reason why a broader application of this material group has so far been restricted despite its high technological potential. Surface machining with lapping kinematics, e.g. lapping, polishing and fine grinding, shows the best performances regarding surface shape and roughness. Nevertheless, these processes show several disadvantages. The conventional lapping process using loose grains takes a relatively long machining time and the subsequent disposal of lapping-suspension is difficult and expensive. A higher chipping rate can be reached by using a fine grinding process with bounded grains, but as the grinding wheels show wear, truing and sharpening of the tool becomes necessary. This is complicated because of the required shape accuracy of the tool and hardness of the abrasive grains [1-4].

A novel machine concept combining the advantages of lapping and fine grinding with lapping kinematics by using foil tools instead of grinding wheels was used to analyze the cutting behaviour of CaF_2 -glass. The first part shows the design of the novel machine concept. The second part deals with the fine grinding process results of CaF_2 -glass.

MACHINE CONCEPT

The design of the novel machine system is based on the machine concept shown in figure 1, which clarifies the constructive requirements. The drafted test rig consists of a plane machined carrier disc combined with a tool drive and output, which provides unused abrasive foil. Those are rotating together with the carrier disc. The workpieces are rotating as well. That way the typical lapping kinematics are accomplished. The grinding force is transmitted by a pressure disc and controlled pneumatically.

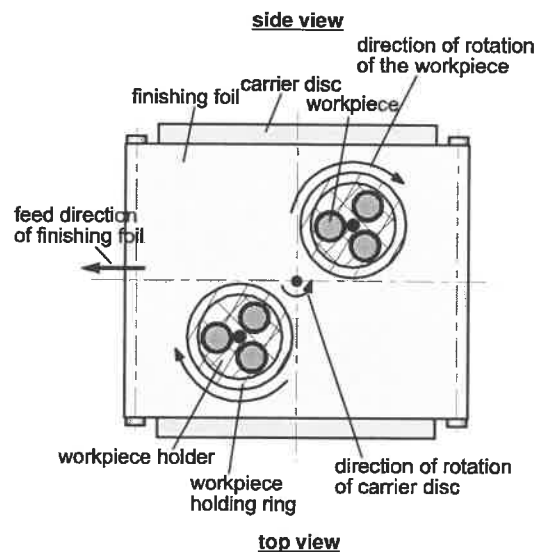
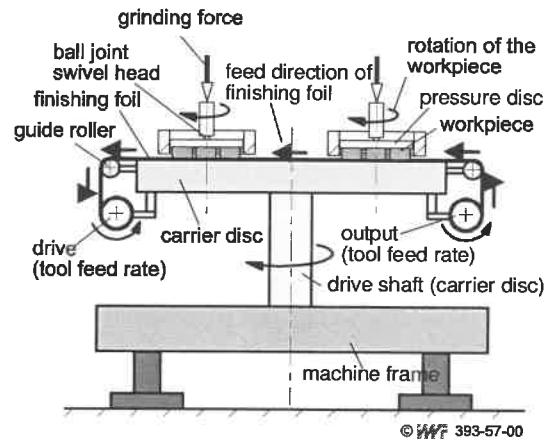


FIGURE 1. Machine concept of the "Fine Grinder".

Fine grinding with feed-controlled abrasive foils eliminates the disadvantages of conventional fine grinding with massive grinding wheels or pellets. These processes require sharpening and truing of the tool and show a non-constant material removal rate. Furthermore, the tool change is complex and time-consuming. The "Fine Grinder" does not have these

disadvantages. The foil drive provides fresh foil and therefore new grains at every times. So, sharpening and trueing are not necessary anymore. Moreover, the process has a constant material removal rate and process behaviour. When the tool-life end is reached, the complete foil-roll is replaced.

TECHNOLOGICAL COHERENCES BETWEEN SETTING PARAMETERS AND MACHINING RESULTS

The influence of foil specification, foil feed rate and cutting speed on the flatness and roughness of the workpiece were investigated.

Due to the brittle material behaviour of CaF_2 -glass a further increase of the grinding pressure over the tested pressure of 0.05 N/mm^2 was not possible. Although the grinding behaviour was more ductile, working with a grinding pressure of 0.1 N/mm^2 already caused a breaking of the workpiece.

Influence of Foil Specification on Machining Results

In order to find ideal foil specifications for the fine grinding process of CaF_2 -glass, four different foil specifications (15 μm , 6 μm and 0.5 μm diamond grain in resin bond, 20 μm diamond grain electro-plated) were examined. The aim was to find a foil specification that provides a good surface quality and a sufficient removal rate. During these experiments the machining parameters were kept constant. Thus, the different foil specifications could be compared.

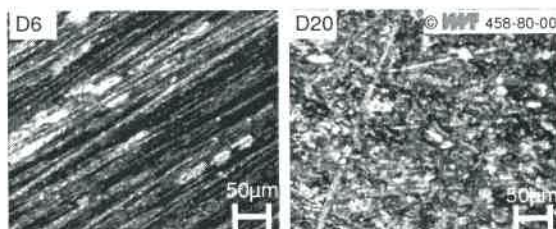


FIGURE 2. Surface of machined CaF_2 -workpiece.

Figure 2 shows the surface of a CaF_2 -workpiece, ground with the abrasive foil D6 and D20. The grooves caused by grinding with the D6 foil are typical for a ductile grinding behaviour. The surface roughness is determined by the crossing of several grooves. The groove depth and in conclusion the roughness depend on the protrusion of the grains. Compared to resin-

bonded grains, electro-plated grains show a higher grain protrusion of the grains and therefore a more aggressive grinding behaviour. In fact, the material removal rate with the electro-plated D 20 foil was 50 times higher than that of the resin-bonded D 15 foil. Moreover, the flatness and surface roughness were not sufficient. It can be seen that the leading cutting mechanism is chipping. This lead to higher strains in the workpiece. Thus, the workpiece broke during machining. In conclusion, the examined electro-plated foil is not suitable for finishing CaF_2 -glass.

Much better surface qualities could be reached by using the resin-bonded foils. The best average surface finish was reached with the D 6 foil ($R_a = 0.03 \mu\text{m}$). Using the D 0.5 foil, the average surface finish improved marginal. The cause for this behaviour is that the material removal rate was extremely low compared to the other foils. In that way, the abrasive foil could not remove enough material to smooth the surface. In addition to that, the fundamental roughness was twice as high as it was for the experiments using D 6 and D 15. Thus, the foil had to remove more material in order to reach an equal roughness.

The choice of foil depends on the requested surface quality. To get the best results several foils could be combined. First, the flatness of the workpiece could be improved within a short period of time by using a D15 foil. Afterwards, the surface roughness could be improved with the D6 foil. The finishing process could be done with the D 0.5 foil.

Influence of Cutting Speed on Machining Results

The surface quality does not only depend on the foil specification but also on process parameters. The influence of the cutting speed on the machining results was investigated using the D 15 resin-bonded foil.

As expected, the cutting speed does not have any influence on the flatness of the surface, whereas the roughness decreases with higher cutting speeds. With a higher cutting speed, the grains enter the grinding zone more often, therefore, more but smaller grooves occur and the roughness decreases. This leads to the assumption that the material removal rate also increases with the cutting speed. However, the experiments showed that the cutting speed has

a contrary effect. The highest material removal rate was achieved with a cutting speed of 0.26 m/s. The material removal rate decreased until a cutting speed of 0.79 m/s was reached and remained static with rising cutting speed (figure 3)

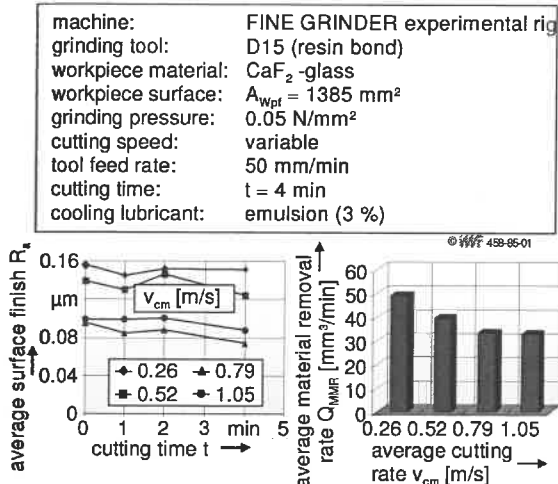


FIGURE 3. Influence of cutting speed on roughness and material removal rate.

The reason for this behaviour lies in the cutting mechanisms of CaF₂-glass. At a low cutting speed of 0.26 m/s the main cutting mechanism is brittle. The grinding behaviour is determined by chipping. In that way the material removal rate is comparatively high as well as the surface roughness. With rising cutting speed, the grinding mechanism becomes more ductile. Thus, the material removal rate declines. Furthermore, the surface roughness is better, since it is determined by the grooves.

Figure 4 shows a macroscopic picture of used D 15 foils at different cutting speeds. At a cutting speed of 0.26 m/s the foil is covered with sediments of CaF₂. This is caused by the high removal rate in addition to the chipping behaviour. Furthermore, the tool wear increases. As can be seen in figure 4, at a cutting speed of 0.26 m/s and 0.52 m/s several diamonds are already pulled out of the resin bond.

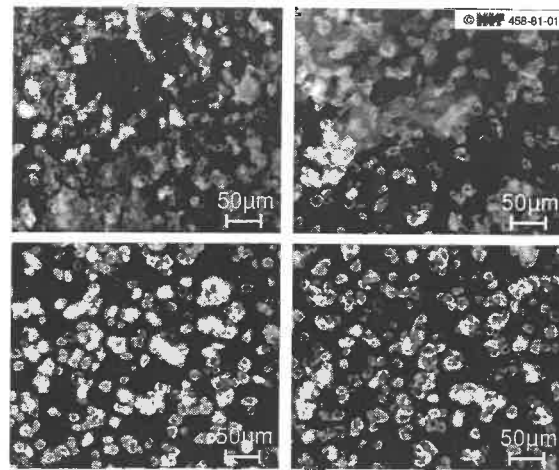


FIGURE 4. Used foils at different cutting speeds.

Influence of Foil Feed Rate on Machining Results

As expected, the foil feed rate does not have any influence on surface roughness and flatness, as can be seen in figure 5. The material removal rate increases with the foil feed rate, since there is a continuous flow of fresh grains into the grinding zone. However, the increasing effect becomes less at higher foil feed rates. At one point the adding of fresh, unused grain is faster than the wear of the abrasive grain. Thus, a further increasing of foil feed rate does not have an effect on the material removal rate.

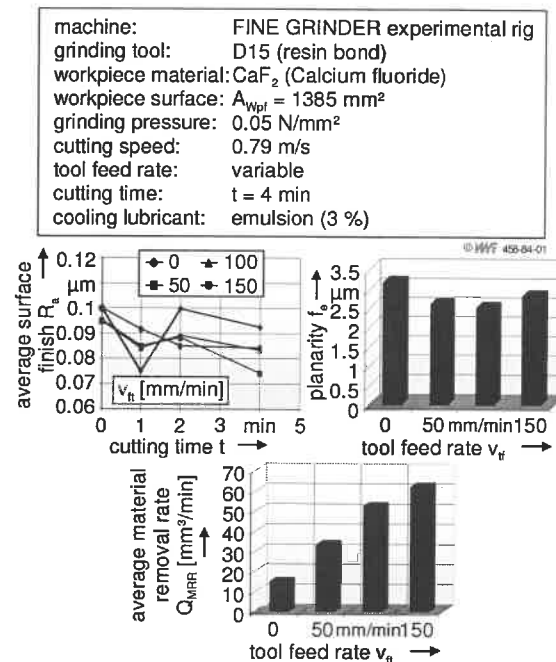


FIGURE 5. Influence of foil feed rate on roughness, flatness and material removal rate.

CONCLUSION

CaF₂-glass is a very brittle material and therefore hard to machine. Small cracks can already lead to a failure of the whole workpiece. A new machine concept using abrasive foils combined with lapping kinematics was used to finish CaF₂-workpieces and the influence of the machining parameters on the finishing results were examined. The highest material removal rate was reached with a D 20 electro-plated foil, but the cutting behaviour was brittle and the workpiece could not resist the aggressive grinding process. The best average surface finish ($R_a=0.03\text{ }\mu\text{m}$) could be reached by using a foil with a grain size of $6\text{ }\mu\text{m}$ in resin bond. Thus a ductile cutting behaviour could be reached. The best flatness of $2\text{ }\mu\text{m}$ could be reached using a D 15 resin-bonded foil, which is almost equivalent to the results of a lapping process. The experiments pointed out that the pressure and the grain size are important for the finishing results, since they determine the cutting behaviour and the strain on the workpiece.

Summed up it can be stated that the new machine concept improves the finishing process of CaF₂-glass. Compared to a lapping or polishing process, the flatness could be improved, since the effect of rounding off the edges did not take place. Furthermore, the attainable roughness is comparable to that reached by a lapping and polishing process. Nevertheless, these good finishing results were reached within a short period of time.

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ELID GRINDING EFFECTS ON FABRICATION OF ADVANCED CERAMICS COMPONENTS

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INSTRUCTIONS

Advanced ceramics are extremely hard to machine using conventional methods. Until now, reproducible mirror surface machining of ceramic material parts has mainly been performed by polishing or lapping. Grinding in particular, compared with lapping and polishing, efficiently produces the geometric form. New grinding techniques for ceramics which employ fine bonded superabrasive wheels and high stiffness grinding machines have been designed to achieve high grinding efficiency and high quality ceramic parts. The ELID grinding technique is a precision machining process involving rigid metallic bonded grinding wheels and a special pulsed electrolytic in-process dressing method [1-7]. In this study, we have conducted the efficient and high precision grinding of some advanced ceramics materials.

ELID GRINDING TECHNOLOGY

Through conducting advanced research on ultraprecision, nanoprecision and ultra-smooth machining processes required for fabricating advanced functional devices, such as optical and electronic components, we have initiated research in a new field of micromechanical fabrication technologies, especially the electrolytic in-process dressing (ELID) grinding technique.

The ELID grinding technique was first proposed by one of the authors. This technique employs a metal-bonded grinding wheel that is electrolytically dressed during the grinding process by continuous abrasives that protrude from the grinding wheel. This enables stable finish grinding with superfine grit to be achieved. The ELID system consists of a wheel bond material, a power supply, and a grinding fluid; the appropriate selection of these components

determines the properties achieved when using ELID.

Figure 1 shows the principle of ELID grinding. The grinding wheel serves as the positive pole, and an electrode attached with a small clearance is used as the negative pole. Electrolysis occurs within this small clearance between the two poles when a chemically soluble grinding fluid and an electrical current are supplied. By generating an insulating layer on the dressed wheel surface and electrolytically removing the bond material during ELID grinding, wheel wear can be reduced by adaptive dressing based on nonlinear electrolysis. Figure 2 shows a schematic illustration of the ELID-grinding mechanism for maintaining ultraprecision finish grinding with fine abrasives.

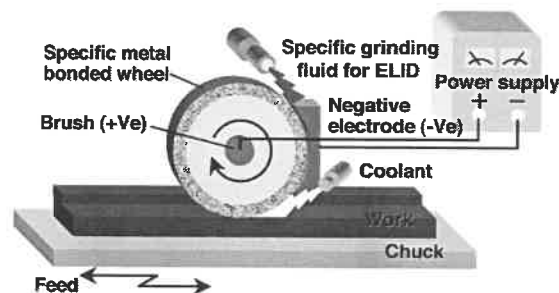


FIGURE 1 Principle of ELID grinding

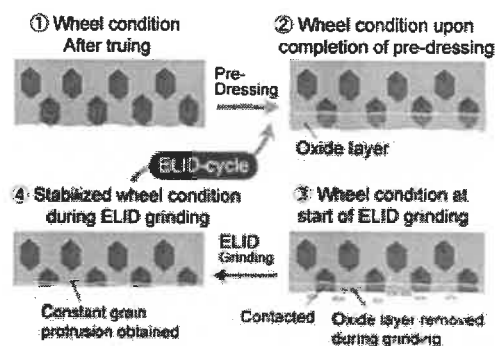


FIGURE 2 ELID-grinding mechanism

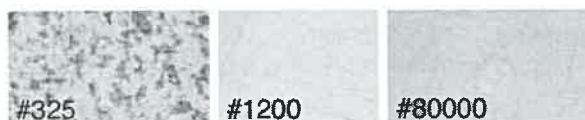
ELID GRINDING OF ADVANCED CERAMICS

In terms of actual research, a variety of parameters were studied in order to achieve a fine surface finish in ductile mode machining. We succeeded in realizing highly efficient, high-precision grinding of the hard and brittle materials such as CVC-SiC (Figure 3(a)), GSO (Figure 3(b)).

We have also strongly promoted highly efficient and precise finishing of a single-crystal SiC wafer that is expected to be used in next-generation power devices. Figures 3(c) and (d) show an image and the surface topography of a SiC surface ground by ELID, respectively. A mirror surface finish was obtained by using a #20000 wheel to grind the SiC wafer. The finished surface roughness P-V and rms values were about 6.2 nm and 0.87 nm respectively.



a) CVC-SiC

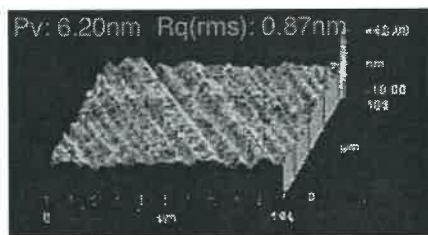


Brittle mode → Ductile mode

b) Ce added Gd₂SiO₅(GSO)



c) ELID ground single-crystal SiC wafer



d) Surface topography of the SiC

FIGURE 3 Samples ground by ELID

As one example of an application we investigated, we performed ELID grinding on aluminum nitride (AlN; used in semiconductor mounting substrates or as a heat-sink material) and evaluated its processing characteristics and surface properties. Figure 4 shows SEM images of surfaces ground using #325 to #30000 grinding wheels. Figure 5 shows the relationship between the mesh size and surface roughness. SEM observations have confirmed that between rough machining using #1200 abrasive and intermediate finishing using #2000 abrasive, aluminum nitride has a brittle-ductile transition point. In addition, the results reveal that final finishing with a #30000 wheel produces an extremely smooth ground surface roughness of 8 nm Ra.

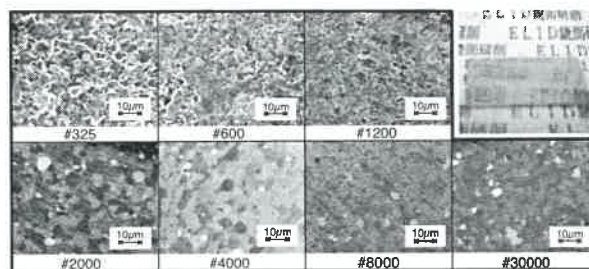


Figure 4 SEM images of surface of AlN ceramics ground by ELID

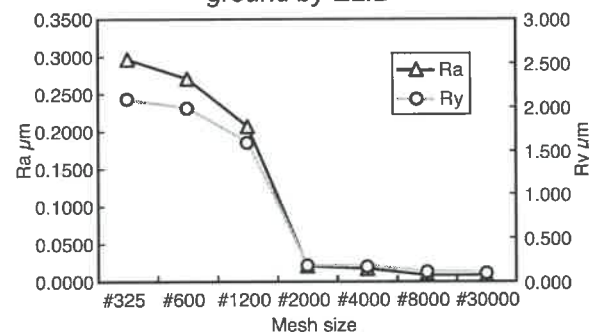


FIGURE 5 Relationship between mesh size and surface roughness

Furthermore, a neutron lens (MgF₂) has been successfully developed. Neutron scattering is essential for investigating the internal structure and dynamics of materials due to the unique nature of neutrons. However, due to the extremely low intensity of neutron beams compared with synchrotron-radiation photoemission, there are limits on its applicability. To rectify this situation, we have been developing a neutron optical device that consists of neutron lenses and neutron prisms. This device consists of an array of several tens of neutron optics, as shown in Figure 6. In this

study, neutron optics were fabricated by applying the ELID technique with #325, #1200 and #4000 cobalt bonding diamond abrasive wheels (Figure 7). Figure 8 shows the neutron lens produced. There is no break on the edge and the groove bottom of Fresnel lens ground by a #4000 abrasive wheel. The Fresnel lens ground by a #4000 abrasive wheel has a very low roughness and its transparency is very high. Also, neutron mirror is being successfully produced by ELID-grinding.

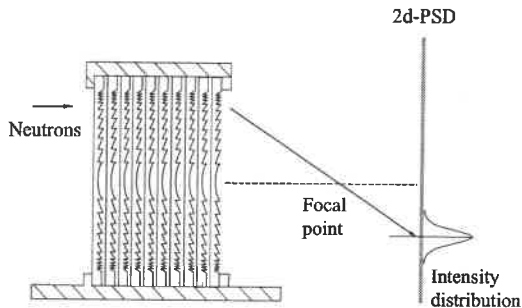


FIGURE 6 Design of a neutron optical device

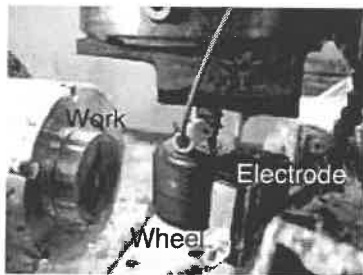


FIGURE 7 Overview of workpiece and wheel set-up

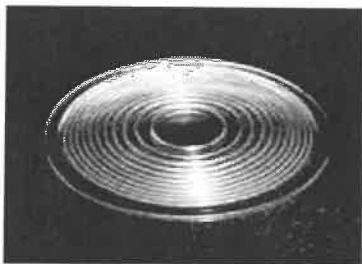


FIGURE 8 Produced neutron lens

More recently, the application of ELID grinding has been tried as a means of achieving high-accuracy machining for next-generation high-power micro chip lasers. Figure 9 shows the process of machining a concave profile with a diameter of 3 mm on the surface of a thin chip of Yb:YAG connected to a heat sink using a #4000 grinding wheel (with an "R" shape tip). Figure 10 shows its appearance after machining. Figure 11

shows the result of measuring the surface roughness; it demonstrates that we succeeded in achieving high quality surface with 4.5 nm PV and 0.45 nm Ra.

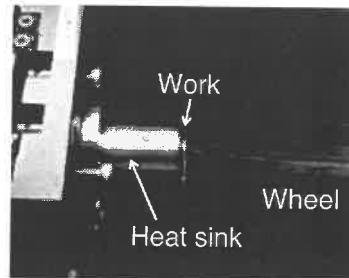


FIGURE 9 Overview of workpiece layout



FIGURE 10 ELID ground micro chip laser

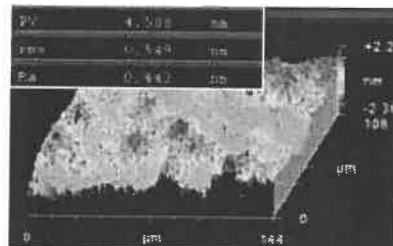


FIGURE 11 Surface topography

Experiments to generate lens mold profile for CVD-SiC ceramics were attempted by using a 4-axes ultraprecision grinding machine. CVD-SiC ceramics is a unique material with superior characteristics in terms of purity (>99.9995%), homogeneity, chemical and oxidation resistance, and thermal and dimensional stability. The wheels used in this experiment were #325 and #1200 for rough grinding, #4000 for intermediate grinding, and #8000 for finishing. In order to improve the form accuracy of desired spherical lens molds, an ultraprecision contact-type compact profile measuring probe (resolution: 2.5nm) was installed on the machine. Thus, fine compensating process has become possible without removal of the lens molds from the machine for measurement (Figure 12). Table 1 shows the surface roughness obtained for different wheel grits. The obtained results indicate that the finer the grain size of the

grinding wheel, the greater the improvement in the surface roughness. A dramatic improvement in the roughness of the ground surface was confirmed between the #1200 and #4000 wheels. This is attributed to changes in the material removal mechanism between the two grains. By final-finish machining using #8000 abrasive, a satisfactory surface roughness of 1.4 nm Ra was obtained. Significant improvement in surface roughness was successfully achieved by fine-grit wheels using the ELID technique.

Figure 13 shows the results of the observation of the ground surfaces obtained with #8000 grinding wheels. It is clear that the surface was stably processed to a smooth surface. A profile after compensation is shown in Figure 14. From this figure, the results confirm that precise lens molds could be processed to high-quality ones at nano levels by the stable performance of both the #8000 grinding wheel having super-fine abrasives and the ultraprecision contact-type compact profile measuring probe.

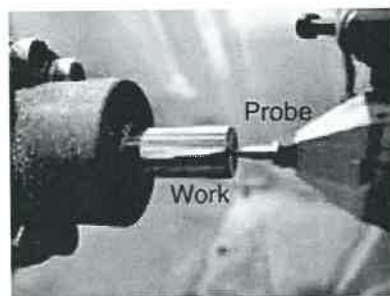


FIGURE 12 Installed ultraprecision contact-type compact profile measuring probe

Table 1 Surface roughness

	PV nm	Ra nm
#325	1894.4	109.3
#1200	1183.0	35.6
#4000	27.7	3.9
#8000	17.1	1.4

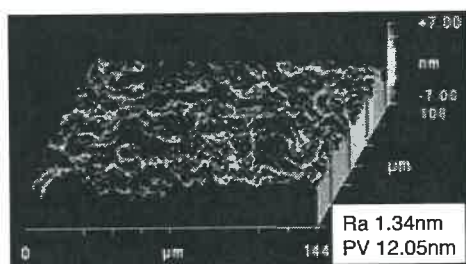


FIGURE 13 Image of ground surface by ELID

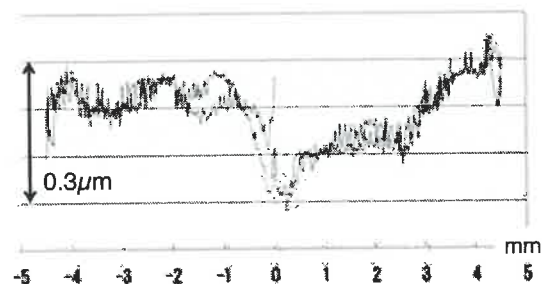


FIGURE 14 Profile by SD#8000 after compensation

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STEADY-STATE DYNAMIC TESTING OF POLISHING PITCH

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SUMMARY

High end systems such as optical metrology, laser and lithography systems all require optical components with tight form and finish requirements. Precision polishing using pitch tooling is regularly used to obtain the part specifications. Pitch tooling consists of a metal platen covered with a layer of pitch. Pitch is a visco-elastic material that is typically derived from pine tree resin. Synthetic, polymer based versions are also available. To date limited data exists on the dynamic properties of polishing pitch and their impact on the polishing process. The work outlined here experimentally measures the steady-state response of pitch samples to forced vibrations of known amplitude and frequency. Pitch samples of varying hardness and thickness were tested. Samples are placed on a shaker table and tested at frequencies varying from 100Hz up to 8kHz. Load cells and accelerometers measure the dynamic response. Initial results demonstrate that below resonant frequencies, neither pitch type or thickness attenuated the source vibrations. Vibrations with nanometer level displacements were easily transmitted through pitch samples as thick as 25.4mm. The geometry of the pitch sample and the mass placed on the sample had a greater impact on the resulting dynamic response than the type of pitch used. These early results suggest that both the geometry of the pitch tool, and the mass of the workpiece being polished will impact the magnitude of the process vibrations transmitted through the tooling to the workpiece.

BACKGROUND

As stated, pitch tooling consists of a metal platen covered with a layer of pitch. Pitch is a viscoelastic material that is typically derived from pine tree resin [1]. Synthetic, polymer based versions are also available [2]. Abrasive particles embedded into the pitch surface during the polishing process enable both material removal and workpiece surface smoothening. Depending on the pitch tool size and pitch grade (hardness), a single pitch tool can have a life

span ranging from two weeks to over a year. Over time pitch tool properties will vary, yet they remain relatively undefined and unmonitored when compared to the tools used in other material removal processes such as grinding.

Current pitch testing is crude and involves either operator experience or a basic indentation test that only captures the materials long term response [1][3]. No instrumentation exists to monitor the condition of the pitch tooling over its life span, and little scientific data exists to direct optimal tooling design. The current success of pitch polishing depends on highly skilled operators with years of experience. It is important to fully understand the pitch's interaction with the polishing process if the process is ever to be deterministic and repeatable. An increased knowledge database on pitch and pitch tooling will result in a more repeatable and cost efficient process. Previous work by conducted by the main author begun to investigate new metrics. Impact frequency response testing was employed and details of how it can differentiate between pitches can be found in the following referenced papers [4][5]. The impact testing revealed that each pitch type (different grades and manufacturers) has its own unique resonant frequency and damping ratios. These results prompted an in-depth look at how pitch behaves under steady state dynamic conditions, i.e. how will it interact with process vibrations. Does pitch amplify or attenuate vibrations? The initial step of investigating the short term, steady state response is outlined here.

EXPERIMENTAL SET UP

Details of the equipment used, the samples tested, and the testing conditions are given in the following sections.

Equipment

Steady state testing of the pitch samples is done on a shaker table. See Figure 1 for a block diagram of the test set up. The shaker table, BK Vibration Exciter Type 4809, can oscillate up to

20kHz. It is driven by a HP Dynamic Signal Analyzer, HP35639A, and BK power amplifier, Type 2706. The dynamic signal amplifier can send both swept and fixed sine waves inputs to the shaker. The force experienced by the vibrating system is measured by an Omega dynamic load cell, DLC101-10 (0-10Lbs). Two PZB accelerometers (1 to 10kHz) measure the accelerations at the base and top of the test sample assembly. A National Instrument DAQ card, with a maximum sampling rate of 500KHz, collects the output data from the load cell and the accelerometers. The signals are recorded using PCScope™. This software package can also perform the double integration on the accelerometer readings to give the corresponding placements (X and Y). Additional data processing is done in MATLAB to calculate the displacement and force amplitudes. Prior to testing pitch samples, the dynamic response of the experimental setup was evaluated with the dynamic signal analyzer. The load cell output was sent to the signal analyzer and evaluated with respect to the driving swept sine wave input (100Hz to 11kHz). The system was found to be linear.

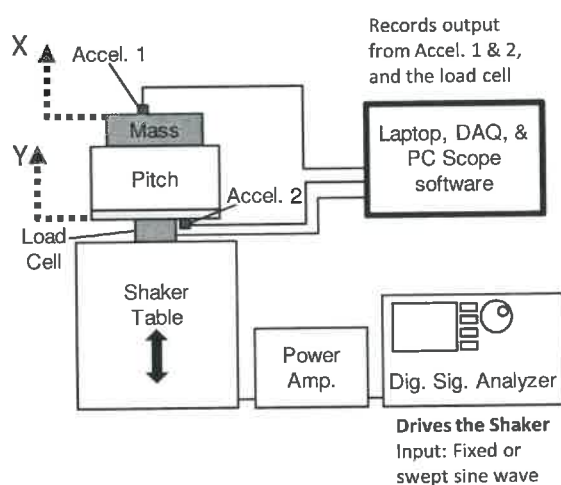


FIGURE 1. Schematic of the main components used in the experimental set up.

Sample Preparation

Two different grades of synthetic polishing pitch were chosen for these initial tests; Acculap™Standard and Acculap™Soft. Two different sample thicknesses were also selected; 25.4mm (mass=50g) and 11.9mm (mass=25g). A 100g mass was attached to the top of the 25.4mm thick sample, while a 125g mass was attached to the top of the 11.9mm thick sample. This was done so that the shaker would exert

comparable forces on the samples tested. To make the samples the interface place, which attaches to the shaker, was placed in the bottom of a silicon mold. A defined mass of pitch was poured onto it to make the thick or thin sample. To pour the pitch it was first heated over a water bath until melted and then heated directly on the hot plate for 5 minutes to make it sufficiently fluid for easy metering into the silicon molds. While the pitch was still warm and compliant, the 'top mass' is embedded in the pitch. Fixturing prevented the top mass from sinking into the sample, or tilting significantly as the pitch cools. This method of sample preparation gave good pitch-metal interface bonding consistencies. See Figure 2 for a schematic of the sample. Samples were let cool over night before testing. Twelve samples in total were made, six thick samples which comprised of three Acculap™Standard and three Acculap™Soft samples, and six thin samples (three of each pitch grade).

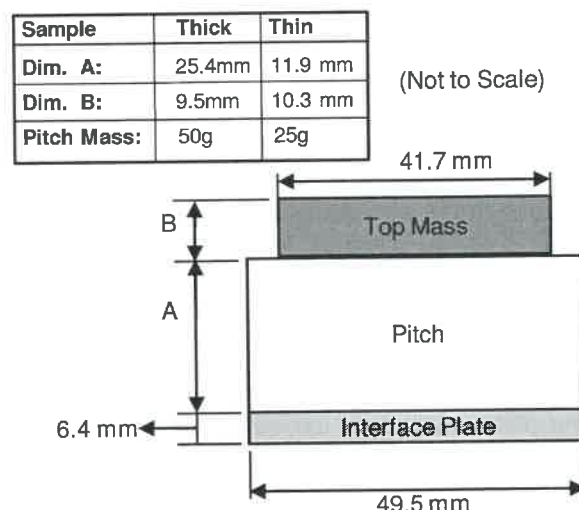


FIGURE 2. Schematics of the samples made for testing.

Testing Conditions

Once mounted on the shaker table each sample was subjected to a swept sine wave (100Hz to 9500Hz). As before the load cell was connected to the dynamic signal analyzer and its response to the input swept sine wave recorded. The resonant frequency of the system (shaker and sample mass) for each sample was determined. As expected the thin samples all have a comparable resonance peak. The same is true for the thick samples. Any significant variation in a samples response from the averaged response implies that there was either misalignment within the sample,

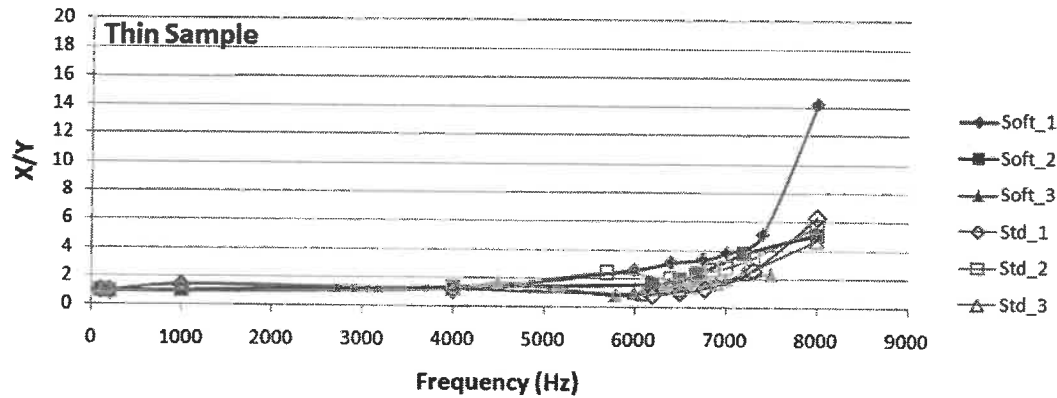


FIGURE 3. Displacement ratio for the thin Acculap™Standard (Std) and Acculap™Soft (Soft) samples.

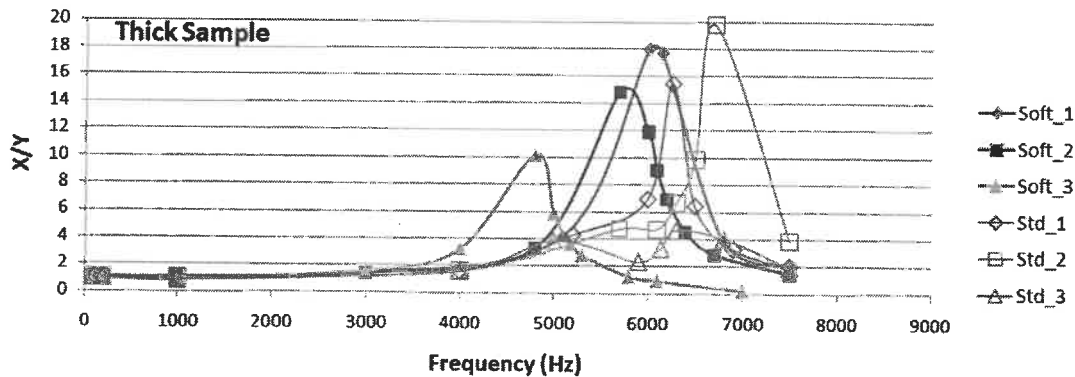


FIGURE 4. Displacement ratio for the thick Acculap™Standard (Std) and Acculap™Soft (Soft) samples.

or that there was poor bonding at one or both of the pitch-metal interfaces. The systems resonant frequencies with the thin and thick samples were $6713\text{Hz} \pm 105\text{Hz}$ (1 std. dev.) and $6118\text{Hz} \pm 167\text{Hz}$ (1 std. dev.) respectively. Once the resonant frequency was isolated, two frequencies on either side of the resonant peak along with the resonant peak were identified for further investigation. Other points included for investigation were 100Hz, 200Hz, 1000Hz, and 3000Hz or 4000Hz. The load cell was reconnected to the DAQ card. A fixed sine wave at each of the above frequencies drove the shaker table and the response of the accelerometers and the load cell recorded. All twelve samples were evaluated with low and high amplitudes inputs. At low amplitude the shaker table displacement ranged from $1.5\mu\text{m}$ to 0.2nm while at high amplitude the displacements ranged from $15\mu\text{m}$ to 2nm .

RESULTS AND DISCUSSION

Figure 3 illustrates the ratio of displacement X to Y measured on the thin samples for different frequencies of the higher amplitude inputs. Figure 4 shows the same information plotted for the thick samples. The same trends as illustrated in Figure 3 and 4 were found when testing at the lower amplitude. Within the bandwidth tested, only the thick samples had a resonance peak.

At lower frequencies ($<4\text{kHz}$), well below any resonance peaks, the displacement ratio between the top and bottom of the samples was approximately equal to one for all samples. The practical implication of this is that neither pitch type nor thickness attenuates the input vibrations. Periodic displacements with peak to valley values as low as 1nm were transmitted through the pitch.

At the higher frequencies ($>4\text{Hz}$), it is clear that the geometry of the samples has more of an impact on the dynamic response of the pitch than the type of pitch used. At resonance (X/Y peaks) the input displacement was amplified up to 19 times as it travels through the thick pitch samples. Within the frequency range tested, the thin samples provided much less vibration amplification. The clear difference in the curves generated by the thin and thick samples is related to either the thickness of the pitch layer, or the change in the mass attached to the pitch. The thinner pitch layer is expected to have a higher associated stiffness than the thicker layer. An increase in stiffness effectively increases the resistance to deformation and the associated resonance frequency of the sample. If testing on the thin samples was continued beyond the 8000Hz , then resonance peaks as seen with the thicker samples should appear. Brown [1], in his paper on polishing pitch, noted that pitch thickness affects polishing outcomes. He focused on tool 'compliance', i.e. the flow of the pitch over several hours, in his explanation of the phenomena. The results presented here indicate that the pitch thickness can impact the shorter term dynamic response. This may also contribute to the differences observed between tools on the workshop floor. Additionally the thinner samples had a larger mass attached to them than the thick samples. The masses may act as passive dampers counteracting and negating system vibrations. Tests are being designed to clarify this point. Irrespectively, the results raise a practical consideration; how does the mass of the workpiece being polished affect the dynamic response of the system? A platen that polishes a large glass workpiece efficiently may not polish a smaller, lighter workpiece of the same material as effectively or indeed vice a versa. Work is continuing in this area.

CONCLUSIONS

Even though these initial tests do not fully explain steady state polishing dynamics, they provide some interesting insights and topics worthy of further consideration.

Below the resonance, neither grade of polishing pitch or sample thickness provided any vibration attenuation. Even vibrations with nanometer level amplitudes were transmitted through the test samples. At resonance frequencies (peaks in Figure 4), both the Soft and Standard pitches amplified the source vibrations. Well above the resonance point (as seen with the Soft_3

sample in Figure 4) attenuation of the vibrations is evident where $X/Y < 1$.

Both the sample geometry and the weight of the top mass, have a greater affect on the dynamic system response than the type of polishing pitch used. This has implications for tool design, i.e. the thickness of the pitch poured on the platen etc. The results also indicate that the dynamic characteristics of a polishing system/tool may not be optimal for all workpiece masses and sizes. Further work is continuing to better understand these considerations.

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RAPID ON-MACHINE MEASUREMENT OF SURFACE FINISH FOR CYLINDRICALLY GRINDING WORKPIECE

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INTRODUCTION

The strict control is required in the surface finish, which is the typical evaluation criterion for finished surface qualities, as well as the shape and size. The flowchart of production process, from grinding (manufacturing) to shipment, is shown in Figure 1. In the actual manufacturing processes by grinding, the post-process measurement of surface finish is carried out after finishing as shown in (a). In addition, the surface finish is generally measured by the surface roughness tester with lower stylus tracing speed of zero point several millimeters per second. Therefore, the surface finish of all parts manufactured in a lot is generally judged based on the measured results of only few parts by the acceptance sampling.

However, if the surface finish measurement is carried out in the grinding process or immediately after the process on the machine tool as shown in (b), the surface finish of all manufactured parts can be checked. It's very useful from the viewpoint of not only quality control of ground parts but also manufacturing efficiency. We propose the novel rapid on-machine measurement of surface finish for cylindrically grinding workpiece and investigate the measurement characteristics.

PRINCIPLE OF SURFACE FINISH MEASUREMENT WITH THERMOELECTRIC EFFECT

The surface finish measurement proposed in this study utilizes the thermoelectric effect as the fundamental principle of thermocouple by which we often measure the temperature. Figure 2 shows the principle of thermocouple and surface finish measurement. The fundamental thermocouple circuit based on thermoelectric (Seebeck) effect is shown in (a). When the temperature at either junction (hot junction) rises in a close circuit composed of a certain metal wire A and wire B of other materials, it will generate a voltage. A pair of different metal wires, for example Chromel / Alumel, Chromel /

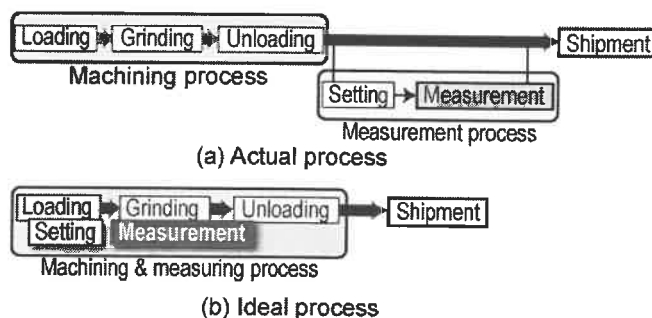


FIGURE 1. Production process with grinding

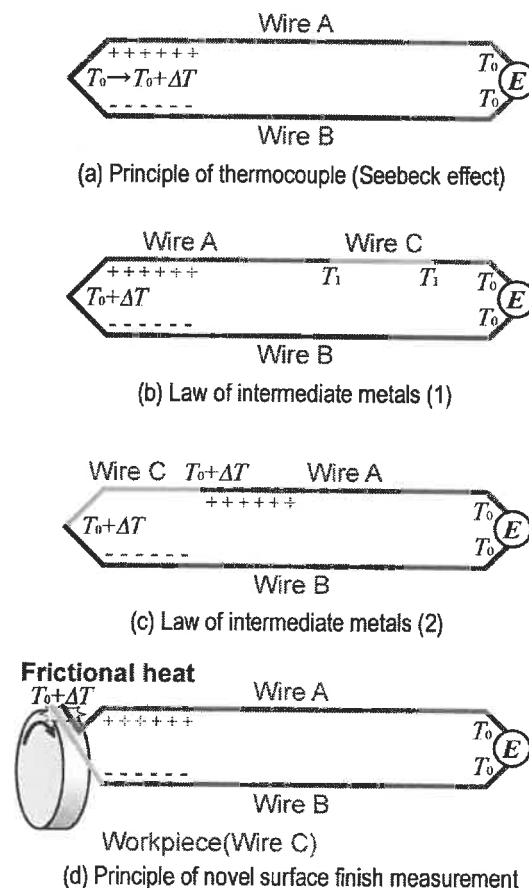


FIGURE 2. Principle of thermocouple and surface finish measurement with thermoelectric effect

Constantan and so on, is often used in the measurement of temperature as thermocouples. When the wire C made of the material different from wires A and B is connected into the circuit with equal temperatures T_1 at both wire ends as shown in (b), the circuit generates the voltage as large as that without wire C [1]. When the wire C places between wire A and wire B as shown in (c), it doesn't affect the generated voltage with the temperature at both wire ends as large as that at the hot junction.

If the ground workpiece is replaced with wire C as shown in (d), the generated voltage reflects the temperature at the points on finished surface contacting with both wires [2]. Furthermore when the workpiece rotates with the contact with both wires on a machine tool, the voltage caused by the frictional heat will be included in that by the temperature of workpiece surface. So if the surface finish has the relationship with the frictional heat with a certain rotating speed of workpiece, the surface finish can be obtained by the generated voltage in thermocouple circuit.

EXPERIMENTAL PROCEDURE

Figure 3 shows the developed rapid measurement system with the surface finish sensor. Chromel and Alumel wires, having no junction each other, are fixed on the sensor body made of acrylic resin with a certain tension, and the sensor makes contact with workpiece surface rotating at about the same rate in grinding process with a constant load. We have an assumption that the workpiece surface is wet during the measurement in or immediately after grinding process, and the coolant is supplied

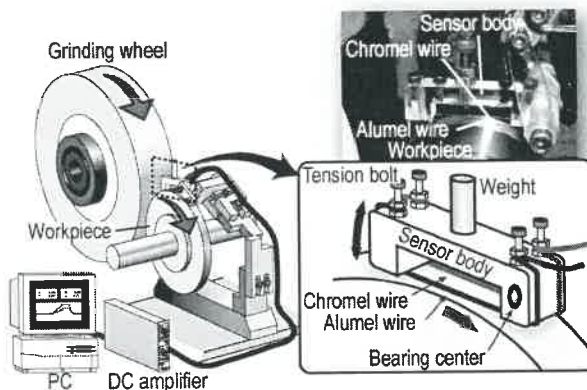


FIGURE 3. Developed rapid measurement system with the surface finish sensor utilizes the thermoelectric effect

TABLE 1. Conditions in surface finish measurement

Workpiece	SUS304 ($\phi 100 \times 10$ mm, Ground surface, $0.2\text{--}1.1 \mu\text{m } R_a$)
Thermocouple	Chromel / Alumel Chromel / Constantan ($\phi 0.32 \mu\text{m}$) Iron / Constantan
Workpiece speed (Measuring speed)	$V_w = 0.16\text{--}0.68$ m/s
Measuring load	$F = 0.19\text{--}0.98$ N
Lubrication	Soluble type coolant (diluted by 1.25%)

onto the workpiece surface during the experimental tests. The sensing signal, the electromotive force (EMF) generated in the thermocouple circuit, is transmitted to PC after amplification. The workpiece surfaces are ground to several surface finishes. Table 1 shows main conditions in surface finish measurement.

RELATIONSHIP BETWEEN SURFACE FINISH AND EMF

Figure 4 shows the variations of electromotive force, which are generated at points on workpiece surface in contact with each thermocouple wire of sensor as shown in Figure 3, when workpiece starts rotating with a certain speed. EMF at room temperature is defined as zero in this paper.

The signals of EMF increase to be stable immediately after turning on the switch of workpiece rotation, and they depend on the surface finish of workpiece R_a . EMF of about $170 \mu\text{V}$ immediately generates by rotation of workpiece with $0.6 \mu\text{m } R_a$ as shown in the figure, and EMF of about $110 \mu\text{V}$ generates by rotation

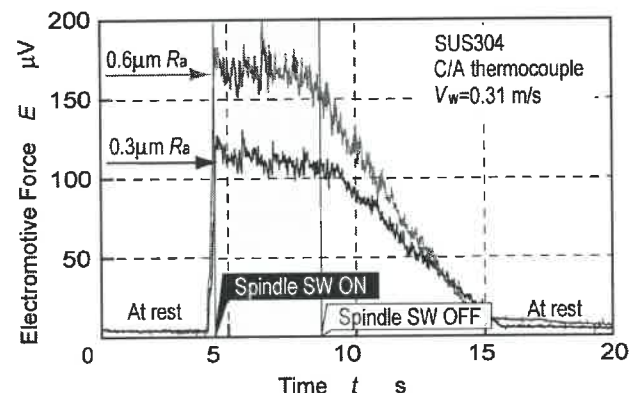


FIGURE 4. Variations of electromotive force EMF in measurement on workpiece surface

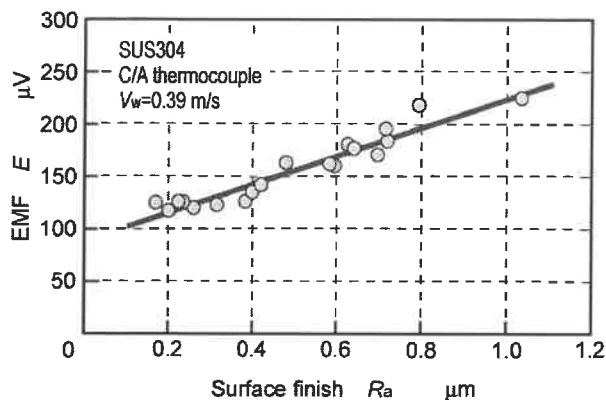


FIGURE 5. Relationship between surface finish and EMF

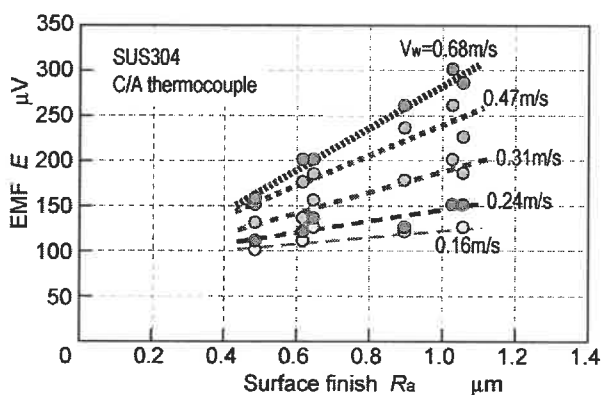


FIGURE 6. Relationship between surface finish and EMF at several measuring speeds

of workpiece with $0.3\mu\text{m}$ R_a . Such difference in EMF suggests the difference in surface roughness. Therefore, the system can immediately obtain the surface finish on grinding machine.

Figure 5 shows the relationship between surface finish and EMF by the measurement system as shown in Figure 3. The arithmetical mean deviation of profile R_a as a surface finish parameter is investigated in follows because it is considered that the sensor output is difficult to reflect very small measured area in the roughness curve differently from the maximum height of profile R_y . EMF approximately has a linear relationships with the surface finish as shown in the figure. This relationship suggests that the frictional heat, generated at the points on workpiece surface in contact with thermocouples wires, increases with an increase of surface finish. Therefore, the surface finish can be estimated by the EMF with the relationship in this figure.

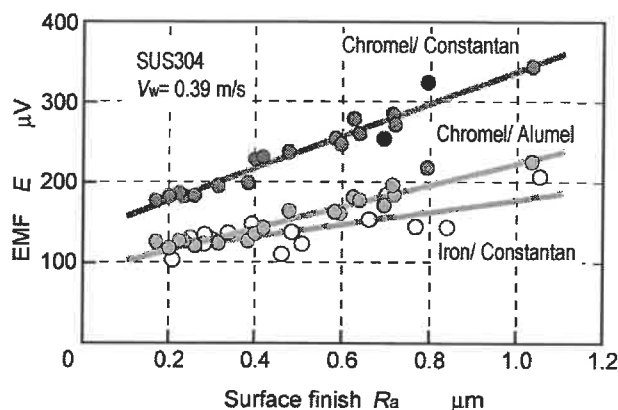


FIGURE 7. Relationship between surface finish and EMF by several thermocouple types

EFFECT OF MEASURING CONDITIONS

Measuring Speed (Peripheral Speed of Workpiece)

Figure 6 shows the relationships between surface finish and EMF at several peripheral speed of workpiece V_w , which is the measuring speed. EMF approximately has linear relationships with surface finish at any measuring speeds and the sensitivity increases with an increase of measuring speed. These relationships suggest that the frictional heat generated at points contact on workpiece surface increases with increasing peripheral speed of workpiece.

Thermocouple types

Figure 7 shows the relationships between surface finish and EMF of three types of thermocouple. EMF approximately increases linearly with increasing surface finish by any thermocouple types like that shown in Figure 5 and the gradient of EMF, which means the sensitivity, is higher in the order, Chromel/Constantan thermocouple, Chromel/Alumel, Iron/Constantan. Therefore, it seems effective to use Chromel/Constantan thermocouple as the developed contact sensor because of its high sensitivity.

Measuring Load

Figure 8 shows the variations of EMF in measurement on workpiece surface, which is ground to $0.5\mu\text{m}$ R_a , at 0.30N and 0.19N in measuring load F . In these measurement processes, the workpiece rotates after setting the sensor on workpiece surface and stops rotating by switch-off of main spindle after steady rotation of workpiece for about 4 seconds.

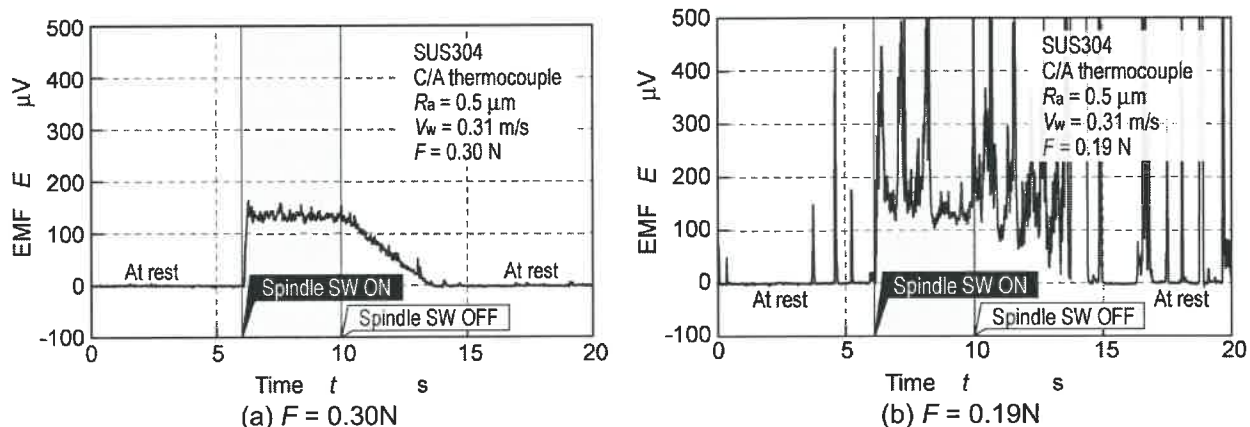


FIGURE 8. Variations of EMF in measurement at 0.30 N and 0.19 N in measuring load

EMF varies so violently that the signal has lots of spikes at 0.19 N as shown in (b). The spikes in signal are also generated after the stop of workpiece rotation. In contrast, EMF immediately increases up to a certain voltage and gradually decreases down to the original amount with decreasing rotation speed of workpiece after switch-off of main spindle as shown in (a). Therefore, it is necessary for this measurement system to apply the measuring load so large that thermocouple wires keep steady contact with workpiece surface.

Figure 9 shows the effect of the measuring load on the sensitivity. The sensitivity at 0.98 N is higher than that at 0.29 N as shown in the figure. Such difference of sensitivity is considered to result from the increase of frictional heat based on frictional force at points on workpiece surface contact with thermocouple wires with an increase of measuring load.

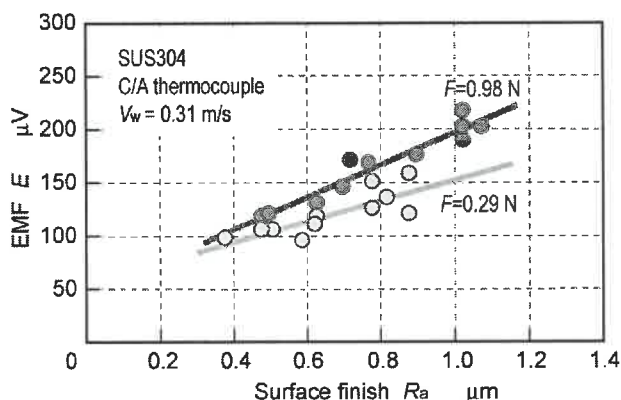


FIGURE 9. Effect of measuring load on sensitivity

CONCLUSIONS

We propose the novel measurement technique of surface finish for rotating ground workpiece with the thermoelectric effect and experimentally investigate its availability and measurement characteristics. Main conclusions obtained in this paper are as follows:

(a) The rapid on-machine measurement technique of surface finish for ground workpiece with thermoelectric effect is developed.

(b) The output from the developed contact sensor with thermocouple wires approximately has the liner relationship with surface finish.

(c) The sensitivity improves with an increase of measuring speed, peripheral speed of workpiece, and the system can obtain the surface finish of workpiece rotating at practical peripheral speed region in grinding.

(d) Chromel/ Constantan thermocouple is best suited to the measurement with high sensitivity.

(e) It is necessary for this measurement system to apply the measuring load so large that thermocouple wires keep steady contact with workpiece surface.

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3D STRUCTURE AND SURFACE TOPOGRAPHY MODELS OF THE MECHANO-CHEMICAL COMPOSITE GRINDING WHEEL BY FRACTAL MODELING TECHNIQUE

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1. INTRODUCTION

In order to realize both abrasion machining with rapid material removal and high dimensional accuracy and interfacial reaction machining with good surface finish and low subsurface damage, we have developed the mechanochemical superfinishing with new vitrified-bonded super-abrasive stones. One stone contains cBN grit and the abrasive cerium oxide, CeO₂ [1], and another contains diamond grit and the abrasive barium sulfate, BaSO₄ [2]. Using these new vitrified-bonded super-abrasive stones for the mechanochemical superfinishing, hereafter referred to as the "mechanochemical composite stones," it was possible to obtain a smooth surface with negative skewness, which could not be produced by using only hard grit stone [1][2]. Clearly, however, it becomes more difficult to decide the production parameters like stone composition, burning temperature, etc. suitable for a given machining performance. Thus, our goal is the development of design system of the mechanochemical composite stone. This paper describes, as the first step of the development, the morphological structure model and the geometric surface model for the cBN-CeO₂ composite stone by using fractal modeling techniques.

2. CONSTRUCTIONS OF MORPHOLOGICAL STRUCTURE MODEL

2.1 Modification of the previous model for the developed one

We proposed two models to represent the structure of a conventional stone, that is, the "agglomerate model of a plastic mixture" representing a plastic mixture before burning it on the basis of the diffusion-limited aggregation

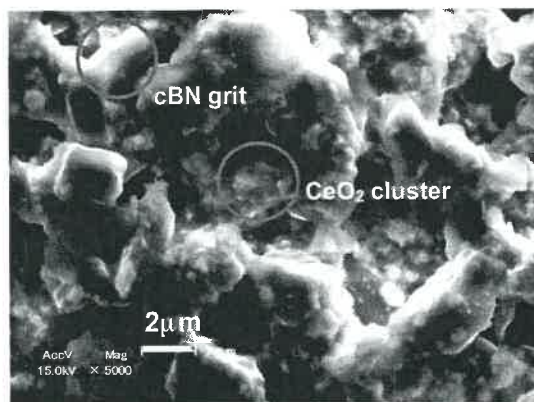
algorithm and the "morphological model of a burnt stone" representing the structure of the burnt stone on the basis of the sandpile algorithm [3]. In comparison with a conventional stone, the characteristics of the cBN-CeO₂ composite stone are as follows: CeO₂ grits tend to agglomerate into some clusters and, since the number of grits bonded to a bond particle is too many, many grits miss being bonded. Thus, in order to incorporate these characteristics to the models, we have modified them by adding the following procedures: the "cluster formation" algorithm and the "judgment of a bonded grit" algorithm.

2.2 Cluster formation algorithm of CeO₂ grits

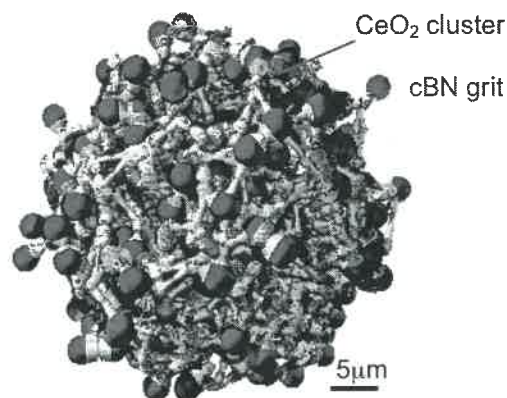
For modeling a plastic mixture of the composite stone, the diffusion-limited aggregation (DLA) algorithm and the simulated annealing (SA) algorithm [3] are applied to the motion of particles of cBN grit, bond, pore agent and "CeO₂ cluster," not CeO₂ grit. The cluster is formed by the cluster formation algorithm and is regarded as a particle. Here, the cluster formation algorithm is the application of DLA and SA algorithms to the motion of only particle of CeO₂ grit.

2.3 Judgment of a bonded grit algorithm

In the burnt stone model of a plastic mixture one formed by the abovementioned algorithm, bond bridges are longer and narrower than those of real stone. Because, in the model, a bond bridge is formed even between grit/cluster particles out of contact each other. Thus, before the application of burnt stone model, the judgment of a bonded grit algorithm is applied to the plastic mixture model. That is, selecting a target grit/cluster particle A, and, when there is no bond particle or cluster connecting particle A and any one of grit/cluster particles in close



(a) SEM micrograph



(b) Morphological structure model

FIGURE 1. SEM micrograph and morphological model of a cBN-CeO₂ composite stone (the weight percentage of material: $w_{\text{cBN}}=34.1\%$, $w_{\text{CeO}_2}=14.6\%$, $w_b=31.7\%$, $w_p=19.6\%$).

proximity to particle A, particle A is eliminated from the mixture.

2.3 Morphological model of a cBN-CeO₂ composite stone

The scanning-electron micrograph and the morphological model of a cBN-CeO₂ composite stone are shown in Fig. 1. As can be seen from this micrograph, some fine CeO₂ grits are attached around one large cBN grit and this characteristic can be expressed in the model. In order to investigate the validity of the model, we compared the box-counting dimension, as a kind of fractal dimension, of the spatial distribution of each abrasive grit in a real stone with that in the model of it. Figure 2 shows the relation between the number of cBN or CeO₂ grit N and the box-size a in a real stone and the model of it. In this figure, a linear relation on a logarithmic graph is observed between a and N as expressed in Equation (1).

$$N \propto a^m \quad (1)$$

where, the exponent m is defined as the box-counting dimension or the fractal dimension. Since the box-counting dimensions m of both grits in the real stone agree well with those in the model as can be seen from this figure, the validity of the model has been confirmed.

3. GEOMETRIC COMPOSITE STONE'S SURFACE MODEL

3.1 Generation of fractal geometric surface

In a cBN-CeO₂ composite stone, the spatial distribution of abrasive grits on the surface has a significant effect on the performance of the mechanochemical superfinishing. Since the 3-D structure of a stone, as mentioned above, is fractal, it is easy to estimate that the surface topography is also fractal. However, the cross

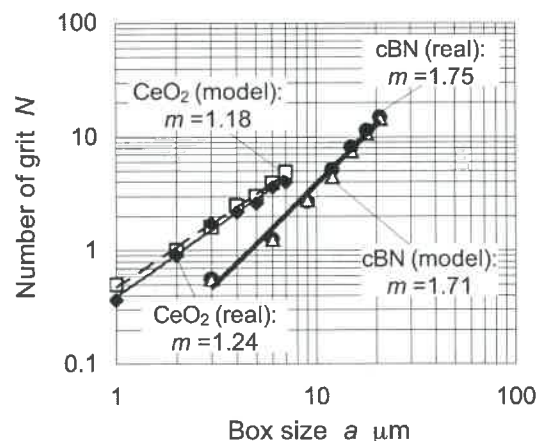
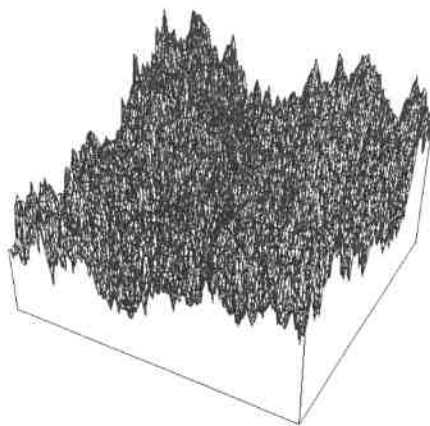
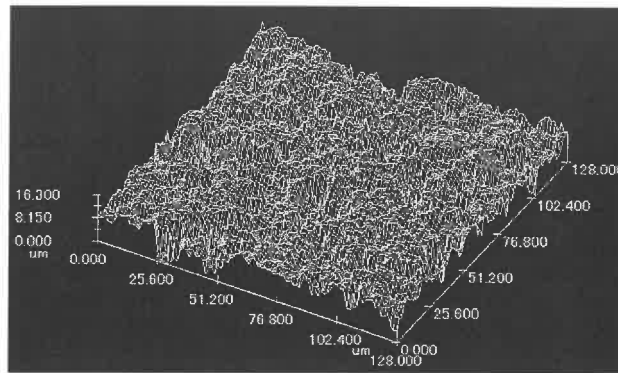


FIGURE2. Relation between the number of cBN or CeO₂ grit N and the box-size a in a real stone or the model of it.

section of the morphological model of a stone cannot be regarded as the geometric surface model of it, because a grit particle is regarded as a sphere. Therefore, the surface model needs to be otherwise constructed separately. We presented the geometric surface model of a conventional stone [4] and this model is generated by the application of the midpoint displacement method and the successive random addition method [5]. Thus, in the model, a square is subdivided into four smaller equal squares and the center point is vertically offset by some random amount based on the fractal dimension of the spatial distribution of abrasive grits on the surface. The process is repeated on the four new squares, and so on, until the desired level of detail is reached. The geometric

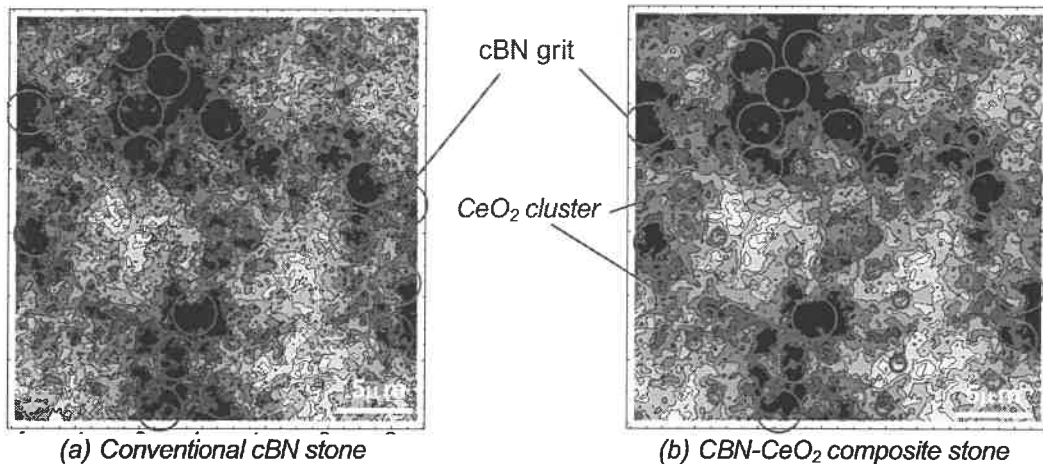


(a) Topography of geometric surface model



(b) Micrograph with LASER microscope

FIGURE3. Topography of geometric surface model and Micrograph with LASER microscope of a cBN-CeO₂ composite stone (the weight percentage of material: $w_{\text{cBN}}=34.1\%$, $w_{\text{CeO}_2}=14.6\%$, $w_b=31.7\%$, $w_p=19.6\%$).



(a) Conventional cBN stone

(b) CBN-CeO₂ composite stone

FIGURE4. Representations with contour lines of the geometric surface models of a cBN-CeO₂ composite stone and a conventional cBN one (model conditions are as follows: Hurst exponent of cBN grit =0.29, Hurst exponent of CeO₂ grit =0.82, variance =30 and seed =15000-25000).

composite stone's surface model is constructed by modifying the presented model.

3.2 Construction of the geometric surface model

Although the previous model, as mentioned above, has been constructed on the basis of the fractal dimension of the spatial distribution of abrasive grits on the surface, the composite stone has two types of abrasive grits such as CeO₂ and cBN grits and both grits have the different fractal dimensions. Thus the modification of the previous model adapted to the composite stone is the following: in the first process to the sixth one, the center point is vertically offset by some random amount based on the fractal dimension of the spatial distribution of cBN grits on the

surface and it is so by that of CeO₂ grits in the seventh process and later.

3.3 Geometric surface model of a cBN-CeO₂ composite stone

The geometric surface model and the micrograph with LASER microscope of a cBN-CeO₂ composite stone are shown in Fig. 3. As can be seen from these subjects, the topography of the model is similar to that of the real stone. However, a cBN grit cannot be distinguished from CeO₂ one in the 3-D representation of this model. Hence the representation with contour lines of the model of a composite stone was compared with that of only cBN one in order to distinguish cBN grits from CeO₂ ones. Here, the model of only cBN stone is constructed under the same conditions of the composite stone

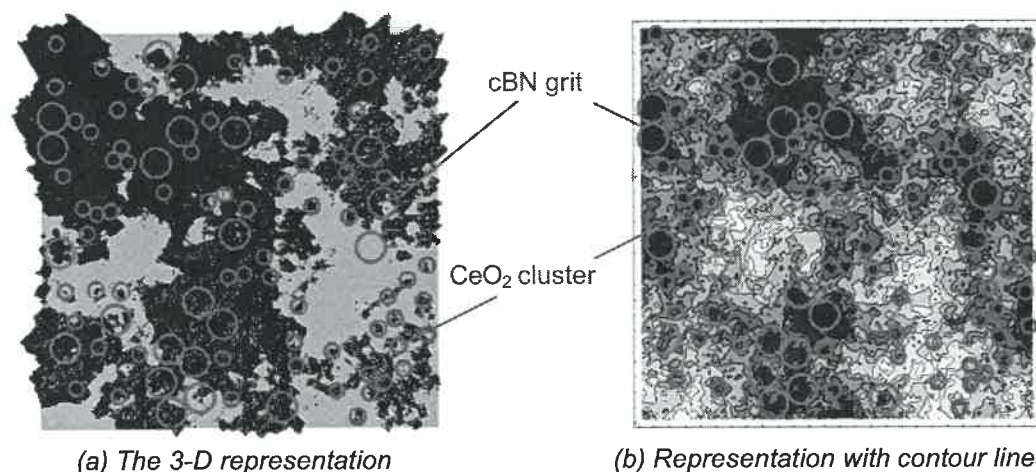


FIGURE 5. Contributions of cBN and CeO₂ grits on the surfaces represented by the 3-D and contour lines (model conditions are as follows: Hurst exponent of cBN grit = 0.29, Hurst exponent of CeO₂ grit = 0.82, variance = 30 and seed = 20000-25000).

except for those concerned with a CeO₂ grit. Figure 4 shows the representations with contour lines of the geometric surface models of a composite stone and a conventional cBN one. As can be seen from this figure, a cBN grit can be distinguished from CeO₂ one by comparing both representations. Moreover, as shown in Fig. 5, this distinction is also possible for the 3-D representation of the model by comparing it with that with contour lines.

4. CONCLUSIONS

We have developed the morphological structure model and the geometric surface model for the cBN-CeO₂ composite stone for the mechanochemical superfinishing of surface, as the first step of the development of design system of it. We developed these models for conventional stones by using fractal modeling techniques. Since the composite stone has two types of abrasive grits differently to conventional ones, we have modified the previous models adapted to the difference. In the morphological structure model, we have added the "cluster formation" algorithm for CeO₂ grits and the "judgment of a bonded grit" algorithm for the burnt stone model. In the geometric surface model, we have modified the modeling process to be adapted to the spatial distribution of two types of abrasive grits with different fractal dimensions on the surface. As a result of the comparison between a real composite stone and the model of it in the micrographs and the fractal dimension analysis, it becomes clear that both models represent the spatial distribution of abrasive grits in the space and surface.

ACKNOWLEDGMENTS

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LEAST SQUARE ESTIMATION OF HARMONIC COMPONENTS IN RADIAL ERROR MEASUREMENT OF A MINIATURIZED MACHINE TOOL SPINDLE

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INTRODUCTION

Miniaturized machine tools have been commonly used for mechanical micro machining of meso/micro scale components. Precise rotation of spindle is essential for accurate positioning of micro cutting tools or work pieces. Measurement and analysis of spindle error motions provides useful insights into the machining accuracy of machine tool [1]. Conventional spindle error measurement methods require multiple capacitive sensors and instrumentations and it is difficult to be implemented in testing the rotational accuracy of miniaturized machine tool spindle due to space limitations. In the present work, single point asynchronous motion (SPAM) test is carried out for evaluating spindle radial accuracy of a miniaturized machine tool using a capacitive sensor and cylindrical artifact in accordance with ANSI/ASME B.89.3.4.M [2]. Figure 1 shows the experimental arrangement of the SPAM test for the miniaturized machine tool spindle.

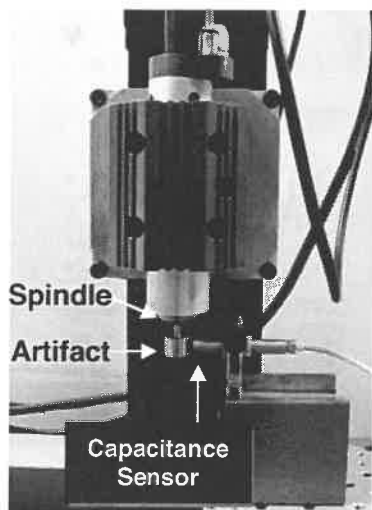


FIGURE 1. Measurement setup for SPAM test.

Surface of the artifact is used as a reference for measuring radial error motions of the spindle. Measurement data is acquired using a computer aided data acquisition system in discrete time

intervals. Sampling frequency of spindle radial error measurement (f_s) is chosen in such a way that it is at least twice the product of desired cycles per revolution (H) and spindle rotational frequency to avoid the aliasing effects [2]. Miniaturized machine tool spindle exhibits speed variations during spindle radial error measurement. Figure 2 shows the measurement data obtained from the capacitive sensor for the spindle speed of 50,000 rpm.

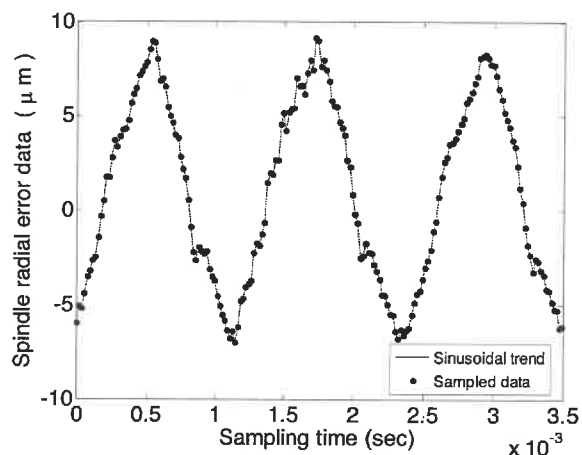


FIGURE 2. Measurement data obtained at the spindle speed of 50,000 rpm.

Measurement data shows a predominant sinusoidal pattern and super imposed harmonic components. It includes the contribution of centering error, form error of artifact and spindle radial error motion. Evaluation of spindle radial error motion requires accurate estimation of harmonic components of measurement data. Prominently used Fourier transform method is not suitable for harmonic analysis of time sampled measurement data [3].

To overcome this difficulty, in the present work, a least square method is used for the harmonic analysis of measurement data. Sum of sinusoidal functions are used for characterizing harmonic components of measurement data.

Experimental results of the proposed method are demonstrated in comparison with Fourier transform method.

LEAST SQUARE METHOD FOR HARMONIC ESTIMATION OF MEASUREMENT DATA

This work extends the best fit sine wave method proposed by Murthy [4], for analyzing the harmonic components of measurement data using a mathematical model consisting of a sum of sinusoidal functions as given by equation (1).

$$m_i = C + \sum_{h=1}^H a_h \cos(2\pi h f_0 t_i) + b_h \sin(2\pi h f_0 t_i) \quad i = 1, 2, 3, \dots, n \quad (1)$$

In equation (1), a_h , b_h represent amplitudes of harmonic components of measurement data. C corresponds to the average position of artifact. f_0 is the fundamental frequency of measurement data. H represents specified harmonic cutoff in cycle per revolution.

If $m_i' = [m_1', m_2', \dots, m_n']^T$ is a set of 'n' samples of measurement data obtained in the SPAM test, harmonic components of measurement data is assumed to follow the equation (1) and it is represented in matrix form as shown in equation (2)

$$m_i' = D x \quad (2)$$

Where D is the matrix containing n rows and $2H+1$ columns as given by equation (3).

$$D = \begin{bmatrix} \cos(2\pi f_0 t_1) & \sin(2\pi f_0 t_1) & \dots & \cos(2\pi H f_0 t_1) & \sin(2\pi H f_0 t_1) & 1 \\ \cos(2\pi f_0 t_2) & \sin(2\pi f_0 t_2) & \dots & \cos(2\pi H f_0 t_2) & \sin(2\pi H f_0 t_2) & 1 \\ \vdots & \vdots & \ddots & \vdots & \vdots & \vdots \\ \cos(2\pi f_0 t_n) & \sin(2\pi f_0 t_n) & \dots & \cos(2\pi H f_0 t_n) & \sin(2\pi H f_0 t_n) & 1 \end{bmatrix} \quad (3)$$

x is the set of model parameters as given by (4)

$$x = [C, a_1, b_1, a_2, b_2, \dots, a_H, b_H]^T \quad (4)$$

Following assumptions are made for estimating the model parameters using least square method.

- Model for estimating the harmonic components of measurement data is true
- Residual errors follow gaussian distribution with zero mean and variance.

Model parameters are estimated by minimizing the sum of squares of residuals (ssr) as given by equation (5).

$$ssr = \sum_{i=1}^m e_i^2 \quad i = 1, 2, \dots, n \quad (5)$$

Here e_i represents the residual error of the fitted model to the measurement data as shown by equation (6).

$$e_i = m_i - \hat{C} - \left(\sum_{h=1}^H \hat{a}_h \cos(2\pi h f_0 t_i) + \hat{b}_h \sin(2\pi h f_0 t_i) \right) \quad (6)$$

Least square solution vector of model parameters for the given value of fundamental frequency (f_0) is obtained using equation (7).

$$\hat{x} = [(D^T D)^{-1} D^T] m_i' = [\hat{C}, \hat{a}_1, \hat{b}_1, \dots, \hat{a}_H, \hat{b}_H] \quad (7)$$

Amplitude and phase of harmonic components can be obtained from equations (8) and (9).

$$\hat{C}_h = \sqrt{\hat{a}_h^2 + \hat{b}_h^2} \quad (8)$$

$$\hat{\gamma}_h = \tan^{-1} \left[-\frac{\hat{b}_h}{\hat{a}_h} \right] \quad (9)$$

It is noted that equation (7) requires the estimation of fundamental frequency of measurement data and it is obtained using following iterative method.

Estimating of fundamental frequency measurement data

Sum of squares of residuals given by equation (5) is written in matrix form using (2) and (3).

$$ssr = (m_i' - D\hat{x})^T (m_i' - D\hat{x}) \quad (10)$$

This criterion can be expressed with respect to $\hat{x}$ and f_0 can be determined using one dimensional grid search method for the maximum of $g(f_0)$ [5, 6].

$$g(f_0) = m_i'^T D [(D^T D)^{-1}] D^T m_i' \quad (11)$$

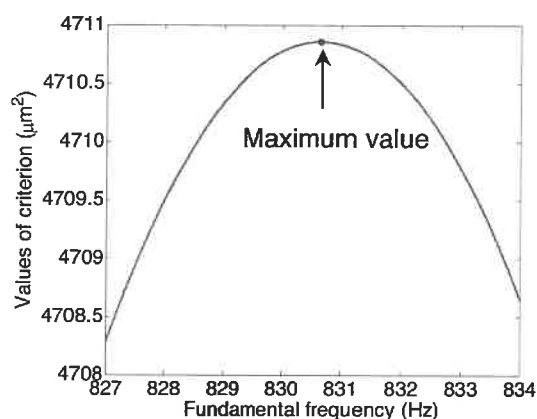
Equation (11) is estimated for the discrete frequency values f_j , $j = 1, 2, \dots, M$ in specified interval $[f_{min}, f_{max}]$ around the spindle rotational frequency (f_r) with a resolution Δf . Grid search iteration method is used to determine fundamental frequency of measurement data that maximizes the criterion given by (11).

EXPERIMENTAL RESULTS

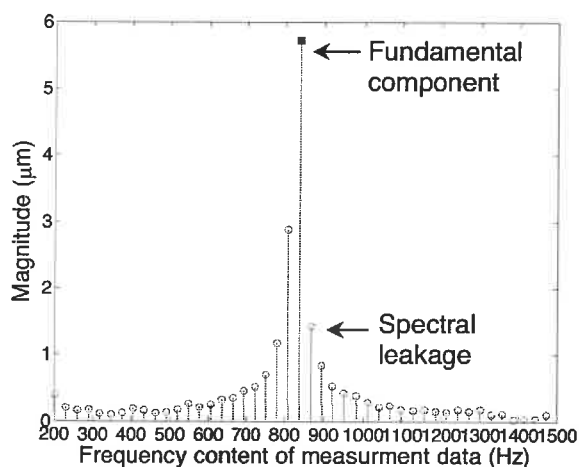
Proposed least square method is applied to the measurement data obtained at various spindle speeds. Harmonic estimation results of proposed method is compared with Fourier transform method. Experimental results are demonstrated for the measurement data obtained at the spindle speed of 50,000 rpm.

Estimation of fundamental frequency

A frequency interval of 827-834 Hz is chosen for determining the fundamental frequency of measurement data, as the spindle rotational frequency (f_r) is 833.33 Hz. Values for the criterion given by equation (11) are obtained at the frequency resolution of 0.01 Hz. Fundamental frequency of measurement data is estimated as 830.69 Hz using grid search method for the maximum value of criterion and the results are shown in Figure 3(a).



(a) Grid search method.



(b) Fourier transform method.

FIGURE 3. Fundamental frequency estimation of measurement data obtained at 50,000 rpm

Figure 3 (b) shows the frequency content of measurement data obtained from Fourier transform method. It exhibits spectral leakage to the adjacent frequency bins due to spindle speed fluctuations.

Estimation of harmonic components of measurement data

Experimental results of least square method for the first 10 harmonic components of measurement data is listed in Table.1 and it is compared with results obtained from Fourier transform method.

Table 1. Harmonic estimation results for measurement data obtained at 50,000 rpm

Number of harmonics	Amplitude of harmonics (μm)	
	Least square	FFT
1	6.82845036	6.77224408
2	0.15301395	0.18002778
3	0.61388373	0.59221204
4	0.07481281	0.07817169
5	0.43201515	0.33542394
6	0.0569406	0.04648886
7	0.18841397	0.11205739
8	0.03663476	0.01845957
9	0.09647231	0.02752655
10	0.04171377	0.02889850

From the results shown in Table. 1, it is observed that Fourier transform method gives smaller values for the amplitude of most of the harmonic components as compared to the least square method, due to the spectral leakage effect. It is also noted that the amplitude of first harmonic or fundamental component of measurement data is dominant among the other harmonic components of measurement data.

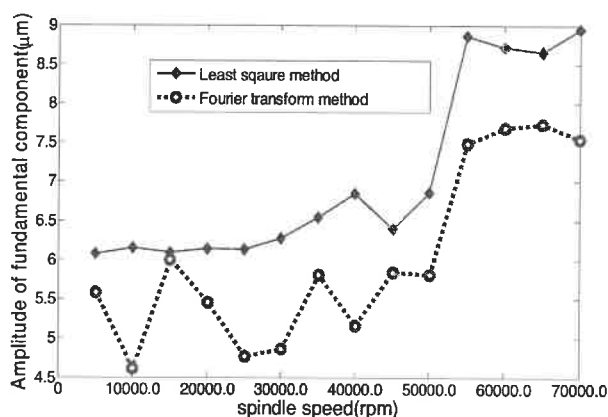


FIGURE 4. Magnitude of fundamental harmonic component for various spindle speeds.

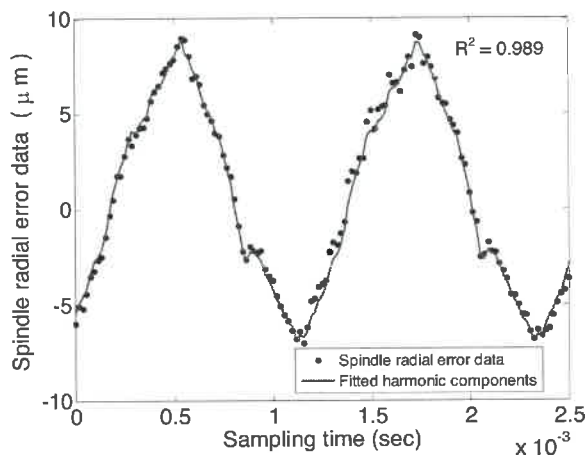
Figure 4 shows the improved estimation results of least square method for the fundamental

harmonic component of measurement data obtained at various spindle speeds.

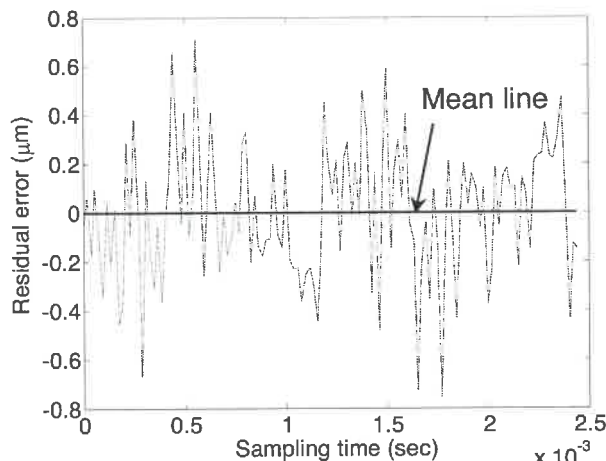
It can be seen that magnitude of fundamental component increases with increase in spindle speed. It is due to the centering error and unbalance mass of the artifact.

Performance of least square fitting method

Performance of proposed method was tested for the short duration measurement data with incomplete number of spindle revolutions and the experimental results are given in Figure 5. It can be seen from Figure 5 (a) that proposed model closely follows the periodic trend of the measurement data. Correlation coefficient (R^2) of the proposed model is found as 0.989.



(a) Least square fitting of proposed model to the measurement data.



(b) Residual errors of measurement data

FIGURE 5. Results of least square method for the measurement data obtained at 50,000 rpm.

Residual errors of measurement data are centered on the mean line without any trend pattern as shown in Figure 5 (b), satisfying the assumptions of least square method. These results prove the suitability of proposed model and least square method for analyzing the harmonic components measurement data.

CONCLUSIONS

Proposed least square method is found to be suitable for the harmonic analysis of spindle radial error data obtained in discrete time interval. Fourier transform method does not provide accurate values for the harmonic components of measurement data due to spectral leakage. The proposed method is suitable for analyzing short duration measurement data with incomplete number of spindle revolutions. The present algorithm requires no frequency domain conversion for the harmonic analysis of measurement data, consequently it can be used for online measurement and analysis of spindle error motions of miniaturized machine tool.

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COMPENSATION FOR THE UNKNOWN GEOMETRY OF A RECONFIGURABLE UNCALIBRATED 3D BALL ARTEFACT FOR FIVE-AXIS MACHINE VOLUMETRIC CHECK

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ABSTRACT

In this study, a mathematical model for removing the effect of the unknown geometry of an uncalibrated artefact for five-axis machine volumetric check is presented. The artefact composed of an adjustable number of master balls is assembled directly on the machine pallet. This design allows flexibility in the number and positions of the balls according to the machine topology and geometry. However, the artefact is uncalibrated as its inherent flexibility hinders precise knowledge of its geometry. A mathematical model is developed in order to assign values to the linear setup errors of each ball and the probe so that their impact on the on-machine artefact probing results can be removed. Tests are conducted in a laboratory on a five-axis horizontal machining centre. The assigned geometry values of the artefact are then verified on a CMM. The experimental results suggest that the mathematical procedure can provide effective values for the compensation of the artefact geometry.

INTRODUCTION

Five-axis machine tools are fundamental elements in the machining of large and complex monolithic parts. Their ability to orientate the cutting tool relative to the workpiece offers a significant reduction in the number of setups required. In addition, manufacturers are now realising that the inspection processes are becoming relatively costly and time-consuming and are looking at ways of reducing their dependence on the part metrology functions while ensuring part geometric and dimensional conformity.

There are different approaches for metrological evaluations of machine tools. One of them is to measure the individual axes by bringing special

instruments within the machine which demands human intervention thus requiring specialised personnel and reducing the machine production time. Another method is on-machine probing of an artefact as a reference part to obtain the machine volumetric status [1-5]. A cube array artefact composed of eight cubes was proposed by Choi et al. [2] to quickly assess the positioning errors of a 3-axis machine tool. The artefact was calibrated on a CMM and then installed on the machine tool. Bringmann and Kung [3] created a pseudo 3D artefact by mounting a 2D ball plate in different locations. They aimed at fast testing and calibration of machine tools, robots and CMMs with three linear axes. Woody et al. [4] developed a technique to transfer the accuracy of a CMM to a machine tool by measuring a part with fiducials both on the CMM and on the machine tool. However, storage and transportation of such artefacts are drawbacks in a production environment.

In this paper, the reconfigurable uncalibrated artefact is described. Then, a mathematical model is proposed for removing the impact of the unknown artefact geometry in the context of five-axis machine volumetric check. Finally, experimental results of the validation of the compensation values for the artefact geometry are presented.

ARTEFACT DESIGN

The artefact is designed to exploit the on-machine probing capability to perform a rapid volumetric assessment of the machine [6]. It is composed of independent (unconnected) master balls mounted at the tips of rods of different lengths forming a 3D artefact. The basic artefact component composed of the assembly of a

master ball, a rod and an insert is presented in FIGURE 1.

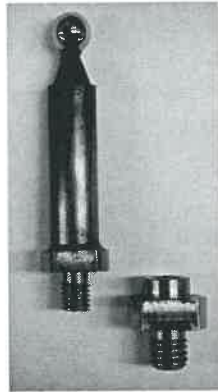


FIGURE 1. Artefact component with an adapter

The new artefact design aims at facilitating its integration within the machine. One of the most important characteristics of the artefact is its reconfigurability. It is made of an adjustable number of master balls located within the machine working and probing envelop, with the rods screwed directly to the standard threaded fixturing holes of the machine pallet. The machine pallet becomes an integral part of the artefact. FIGURE 2 shows an example of an artefact composed of four balls mounted on a five-axis horizontal machine.

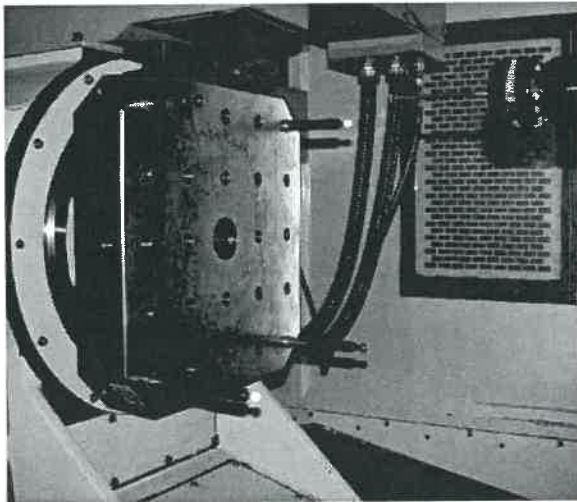


FIGURE 2. An artefact configuration on a five-axis horizontal machine tool of WCBXFZYT topology

MATHEMATICAL MODEL

The artefact is uncalibrated as its inherent design philosophy hinders precise knowledge of

its geometry. A mathematical model is developed to assign values to the linear setup errors of each ball and the probe tip so that their impact on the volumetric errors can be removed.

The actual positions of the master balls and of the stylus tip deviate from their nominal positions by small values called setup errors. As a result, the ball centre data obtained from on-machine probing include not only the effect of the machine errors but also that of the setup errors. A mathematical model is developed in order to assign values to the setup errors in an attempt to explain as much as possible of the measured ball centre data. The model consists of three cartesian setup errors for each of the master balls and another three for the probe. For example, for an artefact composed of four master balls there are 15 error parameters in total.

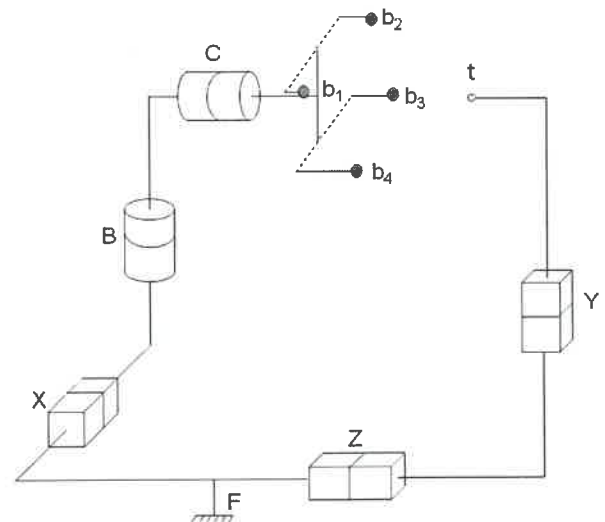


FIGURE 3. Model of a five-axis machine with WCBXFZYT topology

For a five-axis machine with WCBXFZYT topology shown in FIGURE 3, the nominal position of ball b_i relative to the foundation frame can be written as

$$\{F\}_F P_{bi} = \begin{bmatrix} x \\ 0 \\ 0 \end{bmatrix} + R_B R_C \{C\}_C P_{bi}, \quad (1)$$

where x is the nominal motion of axis X, R_B and R_C are the nominal rotation matrices (3x3) of axes B and C, respectively and $\{C\}_C P_{bi}$ is the nominal position of ball b_i relative to axis C.

The effect of the setup errors of ball b_i relative to the foundation frame can be expressed as

$$\{F\}^F \Delta_{b_i} = R_B R_C \begin{bmatrix} \delta_{b_i,x} \\ \delta_{b_i,y} \\ \delta_{b_i,z} \end{bmatrix}, \quad (2)$$

where $\delta_{b_i,x}$, $\delta_{b_i,y}$ and $\delta_{b_i,z}$ are the offsets of ball b_i relative to the C-axis. Similarly for the probe in the tool branch, the nominal stylus tip centre position after the motion of axes Z and Y is given by

$$\{F\}^F P_t = \begin{bmatrix} 0 \\ y \\ z \end{bmatrix} + I_{3 \times 3} \{Y\}^Y P_t, \quad (3)$$

where $\{Y\}^Y P_t$ is the position of the probe relative to axis Y. The effect of the stylus tip setup error relative to the foundation frame is simply

$$\{F\}^F \Delta_t = \begin{bmatrix} \delta_{t,x} \\ \delta_{t,y} \\ \delta_{t,z} \end{bmatrix}. \quad (4)$$

The combined effect of the setup errors of ball b_i and of the probe is data matrix, Δb , which estimates the on-machine probing results (the difference between the nominally calculated positions of the ball centres and those actually measured)

$$\Delta b = \{F\}^F \Delta_{b_i} - \{F\}^F \Delta_t. \quad (5)$$

Substituting Eq.2 and Eq.4 in Eq.5, Eq.6 is obtained in matrix form for a four-ball artefact for one set of axes B and C rotations.

$$\Delta b = \begin{bmatrix} -I_{(3 \times 3)} & R_B R_C & 0_{(3 \times 3)} & 0 & 0 \\ -I & 0 & R_B R_C & 0 & 0 \\ -I & 0 & 0 & R_B R_C & 0_{3 \times 3} \\ -I & 0 & 0 & 0 & R_B R_C \end{bmatrix} \begin{bmatrix} \delta_{t,x} \\ \delta_{t,y} \\ \delta_{t,z} \\ \delta_{b1,x} \\ \delta_{b1,y} \\ \delta_{b1,z} \\ \vdots \\ \delta_{b4,y} \\ \delta_{b4,z} \end{bmatrix} = J_t \cdot \delta p \quad (6)$$

δp is estimated using a least squares criteria.

EXPERIMENTAL RESULTS

A four ball artefact is mounted on a five-axis horizontal machine with WCBXFZYT topology as shown in FIGURE 2 and FIGURE 3. It is measured at seven locations corresponding to B=0°, C=0°, B=30°, C=0°, B=30°, C=60°, B=0°, C=120°, B=0°, C=180°, B=-30°, C=240° and B=-30°, C=300°.

Immediately after the on-machine probing, the artefact is measured on a CMM for validation purposes. The artefact geometry is defined using the distance approach illustrated in FIGURE 4.

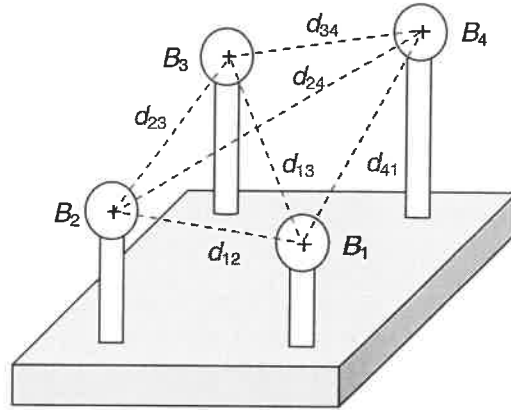


FIGURE 4. Geometry definition of a four ball artefact

The estimated artefact geometry using the previously explained mathematical model and the CMM measured one are compared in TABLE 1.

TABLE 1. Comparison of the estimated artefact geometry with the CMM results

	Estimated [mm]	CMM [mm]	Estimated-CMM [μm]
d_{12}	323.0433	323.0366	6.7
d_{23}	320.0819	320.0829	-1.0
d_{34}	321.9013	321.8944	6.9
d_{41}	329.1863	329.1866	-0.3
d_{13}	456.3643	456.3628	1.5
d_{24}	456.6732	456.6657	7.4

The maximum error between the estimated and the CMM distances is 7.4 μm . The distances are generally above zero which could indicate a scale problem. It is worth noting that the artefact provides no reference length and so the estimated setup errors are probably given in some "machine tool" based composite unit length.

An analysis of the residual volumetric errors, i.e. those that cannot be explained by the setup errors, shows values ranging from 6.2 to 62.5 μm , which indicates that the uncalibrated artefact is able to detect some of the machine volumetric errors.

CONCLUSION

A mathematical method is presented for the geometry determination of an uncalibrated artefact for five-axis machine volumetric check. Tests are performed on a five-axis machine tool. The identified artefact geometry is validated on a CMM. The worst difference between the CMM measured and estimated ball to ball distances is 7.4 μm . These results show that useful geometric setup errors can be assigned to the artefact and stylus tip in order to eliminate their effects from the machine probing results, thus allowing some of the machine volumetric errors to be exposed for machine verification purposes.

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DEVELOPMENT OF OPTICAL SKID SENSOR FOR ACCURACY IMPROVEMENT OF MACHINE TOOL

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INTRODUCTION

Recently, there continues to be ever increasing demands for more precise, complicated and quickly manufacturing. Automation technology by NC machine tool is generally used solve them, but it is difficult to satisfy them at the same time. Especially, trade of is easily generated between precision and another, because of form accuracy of machine tool is dependent of coping principle.

On machine measurement and correction, or controlled, machining system is one of method to solve this problem. Measurement on a machine tool means integration of place for machining and measurement. In the case of measurement without environmental control like machine tool shop, it is influenced of many disturbances, which are thermal, vibration, electric noise, lubricant and so on. Especially, vibration between a workpiece and a sensor cause measurement error.

In this study, the authors propose and develop optical skid sensor to remove the vibration error. Measurement principle chooses triangulation method because it has wide measurement range by comparison with another principle.

OPTICAL SKID METHOD

Displacement measurements by optical probe output averaged displacement in probe spot on measurement surface. In this study, optical skid is achieved by previously described function of optical probe. Optical skid method is composed of a stylus and a skid optical probe.

Since a stylus probe would like to measure surface profile, spot diameter on the workpiece is necessary to reduce less than or equal to several micro meters. If there is displacement error between the workpiece and a sensor caused to vibration or table motion error, output

signal from a stylus probe (St) influenced from the profile and the error. The skid probe measurement profile average because the diameter of the skid probe is more than or equal to several mm. So, since optical signal from the skid probe (Sk) involves only displacement error, $St - Sk$ shows the surface profile.

It is important for high precision measurement that measurement point of stylus is involved of spot area of skid probe on the surface. So, two rays from two light sources are used in many of current optical skid methods. In these methods, it is difficult that two measurement positions at direction of optical axis are adjusted within 0.1 micro meters. If reflecting ray from the workpiece surface is divided in two for skid and stylus, not only adjusting problem is canceled but also the number of optical components, which is influenced of disturbance, is able to be decreased.

In this study, the slit is aligned in front of acceptance surface of the PSD to achieve concurrent measurement of St and Sk. The slit permits the light from centre of the spot to be received at acceptance surface. Figure 1 shows function of the slit in principle of optical skid for triangulation method. In the fact, the slit is aligned in just front of acceptance surface.

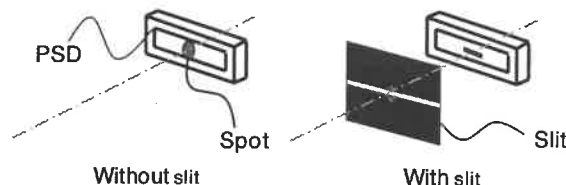


FIGURE 1. Principle of optical skid for triangulation method in case of without slit and with slit.

OPTICAL DESIGN

Figure 2 shows optical system of the sensor. Table 1 shows design specifications of the sensor. The sensor has two PSDs and two Imaging lenses because both S_t and S_k must be measured at the same time. Two acceptance parts are located in opposed position respect to the optical axis of the light source. It makes outputs of the PSDs respect to displacement of measurement surface equal. The PSDs and the lenses are located to implement Scheimpflug condition. In case of a cant surface measurement and a step surface measurement, it is desired for feed direction of triangulation method to square with plane of optic angle.

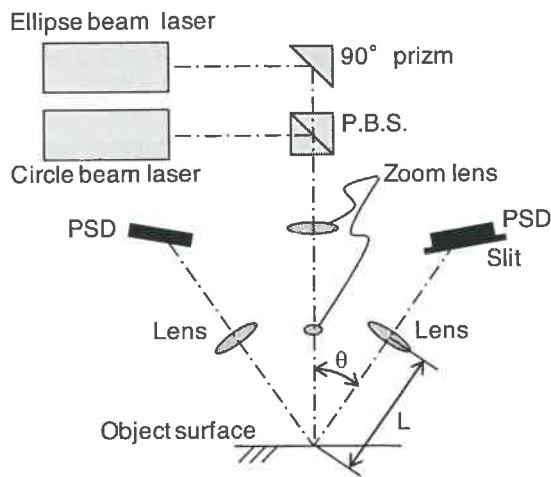


FIGURE 2. Optical system of the sensor

TABLE 1. Design specifications of the sensor

Circle beam size (mm)	$\phi 1.1$
Ellipse beam size (mm)	4.0×1.0
Zoom lens magnification (-)	0.1
Lens focusing length (mm)	18
Lens position L (mm)	36
Optic angle ($^{\circ}$)	35

The measurement range is as 10mm. PSD has the acceptance surface which size is 1×12 mm. The spot on measurement surface is imaged on the acceptance surface by about one-to-one relationship.

S_k has large spot size, which causes measurement error for brightness distribution on acceptance surface, because reflectance distribution is not evenness in the spot on measurement surface. And effect of optical skid is depended on feed direction size of spot. Therefore, the spot shape is used elliptic shape

and long axis is located in the direction of feed. Then the measurement error is decreased as little as possible.

As noted before the slit is located in front of acceptance surface for S_t . Therefore, S_t is depended on slit width relative to S_k is depended on spot size.

EFFECT OF SLIT

To confirm effect of slit, the workpiece which has V-groove shape is measured in several conditions of slit width. Measurement conditions are without slit, slit width $50\mu\text{m}$ and slit width $100\mu\text{m}$. Figure 3 shows measurement result. Figure 4 shows ideal groove shape of workpiece and moving average shape of ideal groove shape in averaging width $100\mu\text{m}$ and $400\mu\text{m}$.

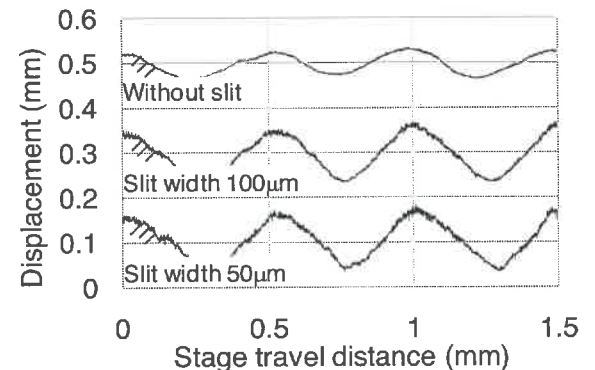


FIGURE 3. Effect of slit for the V-groove measurement

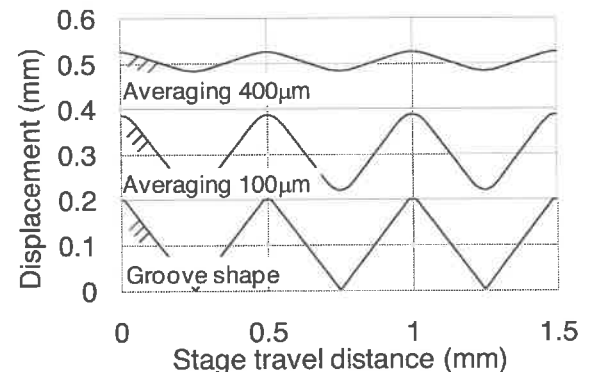


FIGURE 4. the V-groove shape and effect of averaging

Depth of V-groove is 0.2mm . In case of without slit act as S_k , it indicate output amplitude about 0.08mm then averaging effect of S_k is confirmed. This result corresponds to computed result of averaging width $400\mu\text{m}$. output amplitudes in slit width $50\mu\text{m}$ and $100\mu\text{m}$ are about 0.14mm and

about 0.13mm each other. Therefore, to locate the slit in front of acceptance surface makes it possible to extract spot of St from spot of Sk is confirmed. And, confirmed that more slit width is narrowed, the more St is narrowed.

When measurement result compared with calculation result, it shows that measurement result is smoother than calculation result. The cause is considered that scattering light in measurement point is scattered at another surface again and enter in acceptance surface. Because the amount of light received when V-grooves bottom is measured is more than when V-grooves peak is measured.

MEASUREMENT IN VIBRATION ENVIRONMENT

To perform evaluation whether removing effect of vibration is possible or not by the sensor St and Sk are measured at the same time. Then calculation is performed St-Sk to measurement result. Figure 5 shows schema of experimental setup.

The workpiece is same one used before measurement. The workpiece is vibrated by PZT actuator. Feed operation is given by motorize stage in perpendicular direction of plane of paper. In the time of measurement, vibration of reference surface which located in symmetrical position for workpiece is measured by displacement meter to conform vibration condition of measurement point. Experimental conditions are shown in table 2.

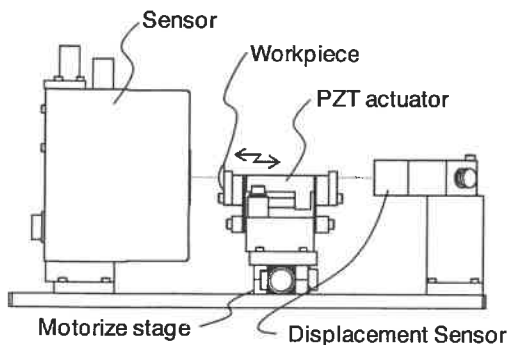


FIGURE 5. Design specifications of sensor

TABLE 2. Design specifications of sensor

Feed rate (mm/sec)	1
Vibration amplitude (μm)	16
Vibration frequency (Hz)	10, 20, 30, 40
Slit width (μm)	50

Figure 6 shows output of St, output of Sk and calculated result St-Sk in without vibration condition. Figure 7 shows same result as previously described and vibration in vibration condition. In the case of focus on figure 7, It shows that both St and Sk are affected by vibration. Output of Sk can be discriminated vibration from shape without difficulty. But output of St can't be discriminated vibration from shape without difficulty. Calculated result of St-Sk is affected by noise, but the result which with vibration and without vibration are corresponded with each other. Sk does not involves only displacement error because smoothing effect is not fully. Therefore, the surface profile is not showed by St-Sk.

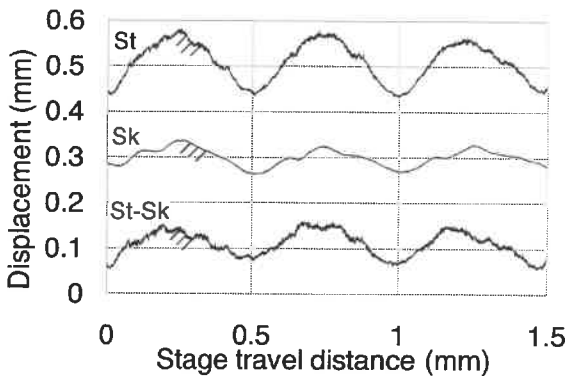


FIGURE 6. Result of V-groove measurement without vibration

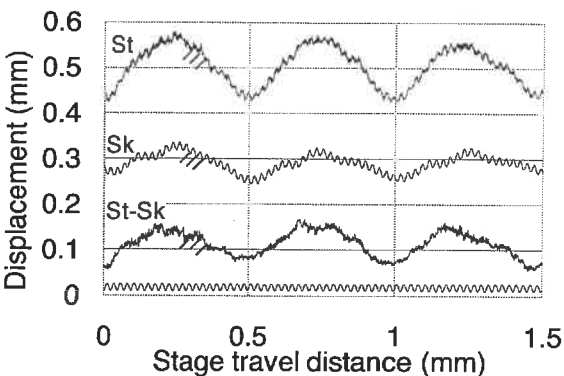


FIGURE 7. Result of V-grooves measurement with vibration (40Hz)

Figure 7 shows RMS and P-V value of error which is difference between St-Sk for without vibration and St-Sk for with vibration of each condition. Measurement in without vibration is measured twice and calculated previously described so that repeatability error by noise is guided. In each case the errors are less than repeatability error. Therefore, it is exhibited that

St and Sk is measured at same time. But it must be needed denoising and advancement of repeat accuracy for high precision on machine measurement.

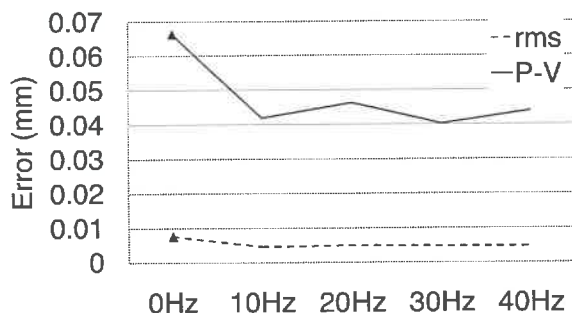


FIGURE 8. Measurement error due to vibration

SMOOTHING EFFECT DUE TO SPOT SIZE

Fully smoothing effect is needed for Sk to only output the shape of workpiece by calculate St-Sk. Smoothing effect is decided by the ratio between spot size (d) and shape size (λ). The more the ratio (d/λ) is larger, the more smoothing effect is larger. And so output change due to spot size change in shape measurement of sine wave is simulated by using moving average. Figure 9 shows simulation outcome. In this simulation spot size is not considered spot diameter and is considered averaging width.

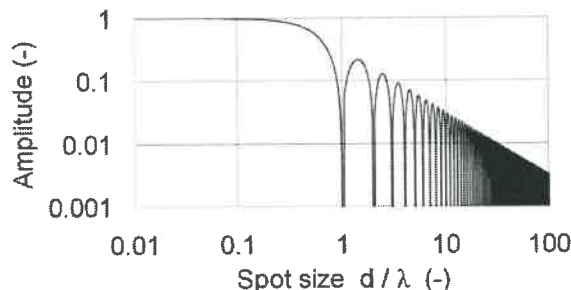


FIGURE 9. Relationship between spot size and output amplitude when constant cycle sine wave shape is measured by several spot size which is defined d/λ .

When spot size is integral multiple of shape cycle, smoothing is performed completely. The more spot size is larger the less output amplitude is decreased. Then the envelope curve which connected the peaks of mountains becomes strait line in logarithmic axis. Therefore smoothing effect is in inverse proportion to spot size.

CONCLUSIONS

In this study, the authors propose and develop optical skid sensor to remove the vibration error which is caused on machine measurement. The prototype sensor is made and the experiment is conducted, and the results of the following are achieved.

- To locate the slit in front of acceptance surface makes it possible to extract spot of St from spot of Sk is confirmed.
- It is exhibited that St and Sk is measured at same time.
- Smoothing effect is in inverse proportion to spot size.

For mentioned above, it is confirmed that effect of vibration can be removed by optical skid method which is advanced in this study.

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SELF-CALIBRATION FOR PRECISION ENCODERS

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INTRODUCTION

Angular encoder accuracy is often the key limiting factor in rotational metrology systems. Inconsistent spacing of the graduations and spindle error motion are major sources of encoder error. Through calibration, the repeatable components of encoder error can be eliminated. Calibration systems have been developed to measure the error map at 128 angular intervals, achieving 0.01" (arc-sec) measurement uncertainty at [1]. Another method using five evenly distributed scanning-heads, self-calibrated graduation errors except 5th order harmonic multiples [2].

This paper presents an encoder self-calibration algorithm for precision rotary stages following original research in [3]. This calibration method can be performed over a large speed range with the encoder on its final axis, and can capture error associated with each position count. Early calibration was performed on an air-bearing spindle with an integrated rotary encoder that was not precise enough to investigate limitations of the algorithm. A precision rotary encoder setup has been designed for this investigation, with a goal of 0.001" measurement uncertainty. To achieve this, the algorithm has been improved to be less sensitive to timing resolution and rotary vibrations.

CALIBRATION THEORY

Most incremental encoders work on the same basic principle. Physical markings on the encoder scale are read by the scanning unit and output as a pair of quadrature signals (A/B) and an index signal indicating the start of each revolution. The rising/falling edges of these signals are treated as spatial sampling events that mark spindle rotary position (FIGURE 1).

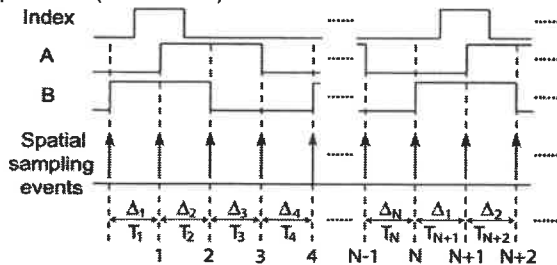


FIGURE 1. Encoder spatial sampling events [3]

Deviations in actual graduation spacing from nominal spacing affect encoder accuracy. Encoder calibration aims to identify these deviations for measurement compensation. On the basis of

earlier work [3], an improved algorithm is developed, as summarized below.

The time at which each spatial sampling event occurs is measured using custom high-speed processing electronics. A dynamic model of the system is used to relate spatial widths to time measurements. The spindle dynamics are modeled as an inertia and damping element, shown below as a differential equation.

$$d\omega/dt + c\omega = 0 \quad (1)$$

The normalized damping coefficient (c , ratio of spindle damping to rotor inertia) is modeled to include constant and speed dependent damping,

$$c = c_0 + c_1(\omega - \omega_0) \quad (2)$$

where, ω_0 , is the initial spindle speed. The angular speed at each sampling event is found by solving a simplified version of the differential equation, using a 2nd-order Taylor series expansion to give:

$$\omega_k = \omega_0 + ak + bk^2 \quad (3)$$

where $a = -c_0\Delta_0$ and $b = c_0c_1(\Delta_0)^2/2$ are damping coefficients estimated below. From the time measurements spindle speed is estimated.

$$\omega_k = \Delta_k/T_k \quad (4)$$

$$\Delta_k = T_k(\omega_0 + ak + bk^2) \quad (5)$$

This equation cannot directly be used for encoder calibration while initial spindle speed and damping are undetermined. Circular closure is used to solve for the initial spindle speed. Dynamic reversal is used to yield a linear equation to solve for damping coefficients:

$$\begin{bmatrix} U_1 & V_1 & -U_2 & -V_2 \end{bmatrix} \begin{bmatrix} a_1 \\ b_1 \\ a_2 \\ b_2 \end{bmatrix} = m_1 - m_2 \quad (6)$$

where U_1 , V_1 , U_2 , V_2 , m_1 and m_2 are vectors containing the time measurement results. A least-square fitting is used to estimate the damping coefficients.

Using (6) damping estimates are overly sensitive to uncertainties caused by limited timing resolution. The proposed method eliminates this sensitivity by using spatial integration to increase the dynamic range of the measurement patterns with respect to time. Instead of equation (6) the modified linear equation becomes,

$$\text{int}\left(\frac{\begin{bmatrix} U_1 & V_1 & -U_2 & -V_2 \end{bmatrix}}{M}\right) \begin{bmatrix} a_1 \\ b_1 \\ a_2 \\ b_2 \end{bmatrix} = \text{int}(m_1 - m_2) \quad (7)$$

where the $\text{int}(\cdot)$ operator represents a cumulative matrix row summation. A least squares fit is

applied and the damping parameters are estimated as:

$$\begin{bmatrix} \bar{a}_1 \\ \bar{b}_1 \\ \bar{a}_2 \\ \bar{b}_2 \end{bmatrix} = (M^T M)^{-1} M^T [\text{int}(m_1 - m_2)] \quad (8)$$

The spatial widths $\bar{\Delta}$ of each encoder count are derived and the encoder error map $\bar{H}(k)$ is found, representing a combination of grating error and radial error motion.

$$\bar{\Delta} = m_1 + \bar{a}_1 U_1 + \bar{b}_1 V_1 \quad (9)$$

$$\bar{H}(k) = \sum_{i=1}^k (\bar{\Delta}_i - \Delta_0) - \sum_{k=1}^N \sum_{i=1}^k \frac{(\bar{\Delta}_i - \Delta_0)}{N} \quad (10)$$

SIMULATION RESULTS

Simulation is used to evaluate the theoretical accuracy limit of the calibration algorithm. Precise time measurements are generated at each encoder count using the simulated spindle dynamics from (1), experimental damping coefficients and a reference error map. The time measurements are quantized to a timing resolution of 1.2GHz, to simulate the high speed processing electronics in the experimental setup. The simulated time measurements are similar to those generated from the experimental setup but only reflect timing uncertainty. Self-calibration is performed on the simulated time measurements and a new calibrated error map is found. The accuracy of the method is found as the rms difference between the reference and calibrated error maps. The simulated accuracy of self-calibration is expected to follow time quantization error, and increases with the speed of rotation. Accuracy of self-calibration with and without spatial integration is compared for a speed range from 10 to 1000rpm (FIGURE 2). Spatial integration is necessary for accurate system identification. This ensures the uncertainty introduced due to limited timing resolution is minimized and the theoretical limit is achieved.

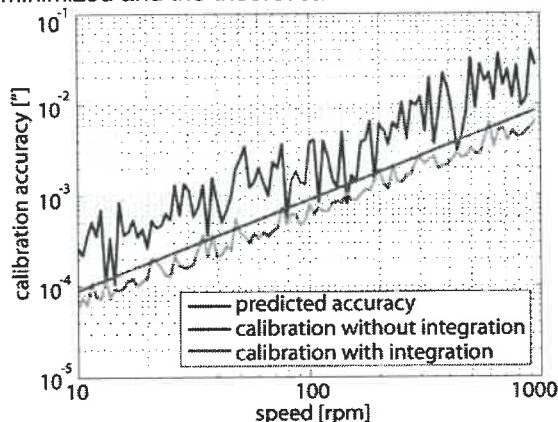


FIGURE 2. Simulated self-calibration accuracy.

EXPERIMENTAL SETUP

A precision rotary table, high-speed electronics and software have been developed to experimentally

investigate the limits of the self-calibration algorithm [4]. The precision rotary table consists of an air-bearing spindle (Professional Instruments 10R block-head), and a rotary encoder (Heidenhain 4282C), shown in FIGURE 3. The rotary encoder has 32768 graduation lines, 131072 counts/rev and specified system accuracy of $\pm 2''$.

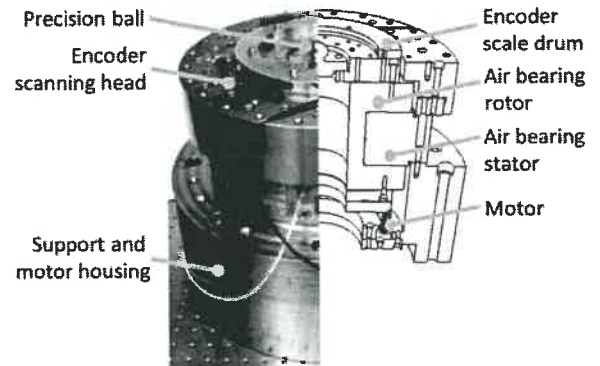


FIGURE 3: Precision encoder experimental setup.

The setup can accommodate up to four encoder scanning heads spaced 90° apart from each other (FIGURE 4). Alignment of each read head is ensured to sub-count accuracy ($\pm 10''$) by checking cross-correlation of error maps and aligning phases on the quadrature signals between each head. The customized electronics have been designed for fast and accurate time measurement and signal interpolation and is capable of capturing encoder pulses at a 1.2GHz clock rate. The software can be used to continuously log time data, allowing calibration over a large speed range.

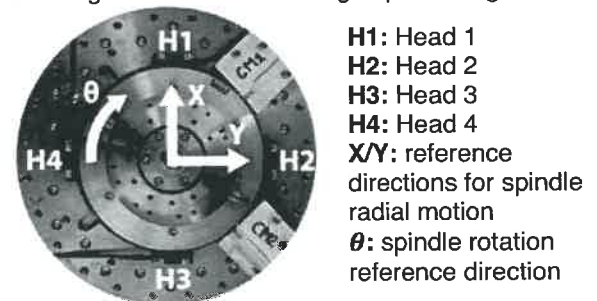


FIGURE 4: Top view of the encoder setup.

SINGLE HEAD SELF-CALIBRATION

Self-calibration was first investigated using a single encoder scanning head in position H1. FIGURE 5 shows the encoder error map obtained at 270rpm with 500 harmonic components included. The error map is dominated by a 1cpr component, largely influenced by graduation scale eccentricity, which is only captured with on-axis calibration methods. The experimental repeatability of calibration is determined as the rms difference between two error maps derived sequentially. Repeatability measurements show the same pattern as the

simulated results, which are proportional to speed due to increasing timing uncertainty (FIGURE 6). Experimental repeatability measurements do match the predicted simulation accuracy due to limited signal to noise ratio on the encoder signal causing additional uncertainty.

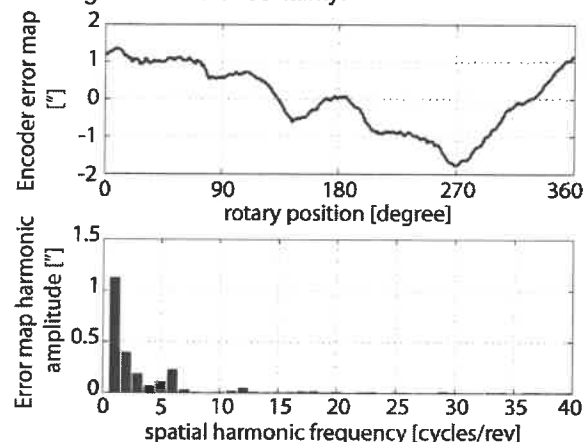


FIGURE 5: Single head calibration at 270rpm. Encoder error with 500cpr harmonics included (top), error map harmonic amplitude (bottom).

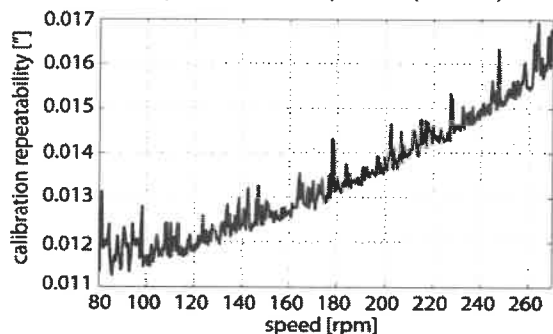


FIGURE 6: Calibration repeatability

FIGURE 7 shows the first 6 harmonic components of error maps across speed, by continuous time measurement and calibration. The amplitudes of error map harmonics change significantly with speed. This was identified as rotary vibration caused by disturbance torque during the spindle free response. This vibration, not modeled in the self-calibration algorithm, can introduce error into the calibration. As a result, the calibrated error map from (10) for encoder head H1 contains not only grating error $G(\theta)$ and radial error motion $Y(\theta)$ components, but also rotary vibration $V(k)$. The error map can be modeled as

$$H1(\theta) = G(\theta) + Y(\theta) + V(\theta) \quad (11)$$

The rotary vibration component can be compensated using a least-square fitting on individual error map harmonics assuming grating error and radial error motion are constant and rotary vibration is proportional to the inverse of speed squared.

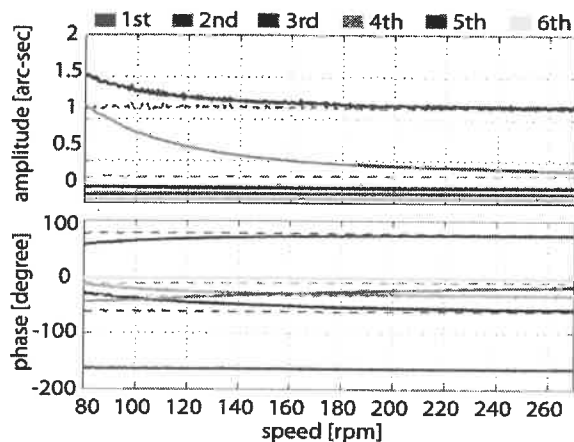


FIGURE 7: Results of head H1 error map harmonics (grating error, radial error motion and rotary vibration). Measured (solid line) and vibration compensated (dashed)

MULTIPLE HEAD COMPARISON

To more accurately remove rotary vibration from the calibrated error map, a multiple scanning head comparison method was developed. Each calibrated error map from individual scanning heads is composed of a unique combination of grating error, radial error motion along X or Y and rotary vibration.

$$\begin{aligned} H1(\theta) &= G(\theta) & +Y(\theta) & +V(\theta) \\ H2(\theta) &= G(\theta - \pi/2) & -X(\theta) & +V(\theta) \\ H3(\theta) &= G(\theta - \pi) & -Y(\theta) & +V(\theta) \\ H4(\theta) &= G(\theta - 3\pi/2) & +X(\theta) & +V(\theta) \end{aligned} \quad (12)$$

The grating error component rotates with head position, error motion along X or Y will only effect measurement when orthogonal to the head and rotary vibration is rotor position dependent and the only component identical among heads. FIGURE 8 shows error maps from four heads calibrated at 100 rpm.

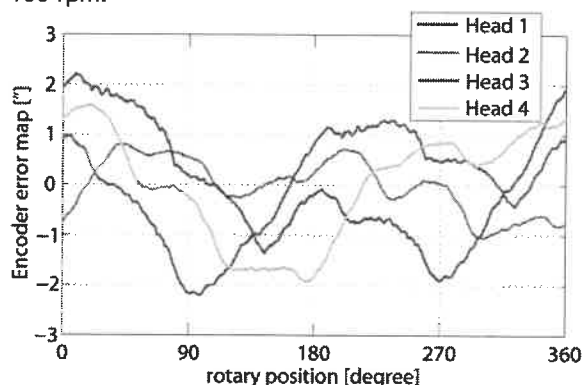


FIGURE 8: Individual scanning head error maps at 100rpm

Each component of the calibrated error map can be isolated using combinations of measurements from the four heads. To remove rotary vibration, error maps from each scanning head are summed,

cancelling radial error motion of opposing heads and grating error harmonics. The extracted rotary vibration component at 100rpm is shown in FIGURE 9. FIGURE 10 shows the rotary vibration over the speed range where a -40dB/decade slope is observed. This disturbance torque doesn't change with spindle speed, but depends only on the rotary position.

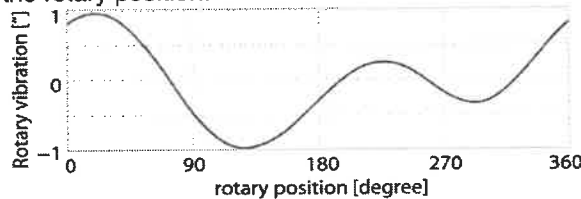


FIGURE 9: Extracted rotary vibration at 100rpm.

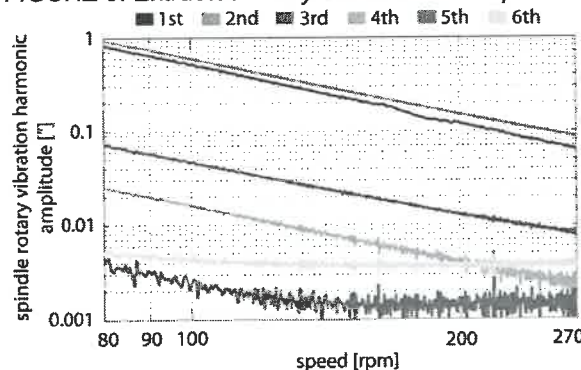


FIGURE 10: Extracted rotary vibration over 80 to 270rpm speed range.

The multiple head comparison method allows rotary vibration to be identified and eliminated from the measurement. FIGURE 11 shows the compensated error map harmonics of head H1 by removing the spindle rotary vibration components, constant over a large speed range. Results from this analysis match estimation performed on single head error maps harmonics in FIGURE 7.

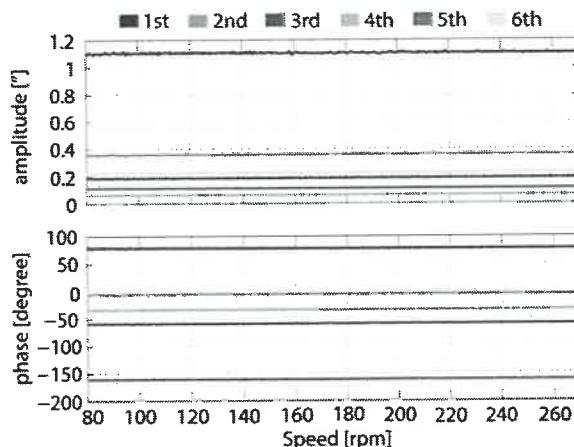


FIGURE 11: Compensated error map harmonics for head H1 (grating error and radial error motion)

Compensated error maps for 100rpm are shown in the top graph of Figure 12. The error map variation from 100 to 200rpm is less than 0.01" rms (bottom graph of FIGURE 12). For better precision, four encoder heads can be averaged to get more accurate spindle angle measurement. The uncertainty of averaged measurements between 100 rpm and 200 rpm is less than 0.005" ($k=2$).

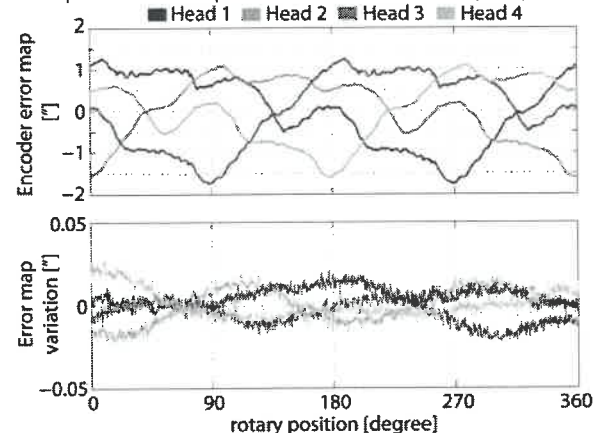


FIGURE 12: Compensated error map of all four heads.

CONCLUSION

A novel self-calibration algorithm for precision rotary encoders has been developed that uses a spatial integration technique to improve the accuracy of inertia and damping coefficient estimations. A precision rotary table setup was used to investigate the performance limits of the algorithm. A rotary vibration caused by disturbance torque, was identified from encoder calibration results. A method using multiple scanning heads was developed to extract the rotary vibration from the error map and eliminate it. Compensated error maps are repeatable within 0.01" for single head and 0.005" ($k=2$) for four head averaging, compared at spindle speeds of 100 and 200rpm.

ACKNOWLEDGEMENT

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EVALUATION METHOD FOR MULTI-AXIS ERRORS OF A 4 AXIS MACHINE TOOL

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INSTRUCTIONS

This paper describes a method to measure and compensate a 4 axes ultra-precision machine tool that generates micro patterns on the large surfaces. The grooving machine is usually used for making a micro mold for many electrical parts such as a light guide plate for LCD and fuel cells. The ultra precision machine tool has three linear axes and one rotational table. Shaping is usually used to generate micro patterns. In the case of 50 μm pitch and 25 μm height pyramid pattern machining with a 90° wedge angle bite, one of linear axis is used for long stroke motion for high cutting speed and other linear axis are used for feeding. The triangular patterns can be generated with many times of long stroke of one axis. Then 90° rotation of work piece is needed to make pyramid patterns with superposition of machined two triangular patterns.

To make a two dimensional positioning error,

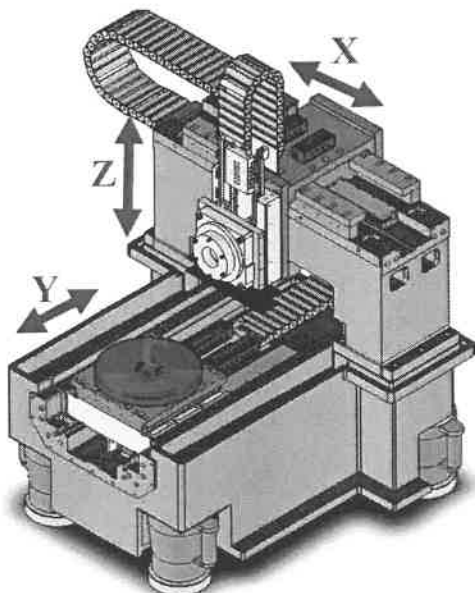


FIGURE 1. Configuration of the 4 axis ultra precision machine

straightness of two axes in out of plane, squareness between the each axis are important. Positioning errors, straightness and squareness were measured by laser interferometer system. Those were compensated and confirmed by ISO230-6. One of difficult problem to measure the error motions is squareness or parallelism of axis between the rotational table and linear axis. It was investigated by simultaneous moving of rotary table and XY axes. This compensation method is introduced in this paper.

CONFIGURATIONS OF THE MACHINE TOOL

Figure 1 shows the basic configurations of the ultra-precision 4 axis machine tools. It has 3 linear axes and 1 rotational axis to be used in shaping, milling and turning process. The hydrostatic oil bearings are used for the guide of all the 4 axes, regarding compactness and high stiffness of the machine tools. To maintain non-contact state, the coreless linear motors are used for actuators of 3 linear axes and a frameless rotating motor is inserted in a rotating table.

The machining volume of the machine tool is 500 mm $\times$ 500 mm $\times$ 150 mm and the rotating axis (c-axis) is inserted in Y axis. The base of the machine tools is made from cast iron and supported by four air-springs vibration mount. The vertically moving axis (Z-axis) weighs 70.8 kg and wiring with an another 70.8 kg counter mass and mounted on the X axis. The position sensor of the each linear axes is a hologram precise optical scales, which has 650 μm precise grating and the resolution is changeable from 300 nm to 1 nm.

EVALUATION AND COMPENSATION OF THE LINEAR AXES

The error of the z axis; small moving axis; is relatively smaller than X and Y axes, because the machine tool is used for generating micro

patterns. Therefore, the 2-dimensional position errors are main errors to be compensated. The two dimensional (2D) errors are occurred owing to relative motion of the two axes. The 2D position errors on the XY plane are mainly affected by straightness, pitch, yaw and perpendicular errors of the each axis. In the case of machining micro patterns on the plane, the change of height along to Z axis is less than 100 μm and effect of yaw error can be simplified by assuming tool point is on the center. Therefore, the measured 1D position error can replace the errors owing to yaw and pitch errors and can be expressed by

$$\begin{bmatrix} \delta P_x(i, j) \\ \delta P_y(i, j) \end{bmatrix} = \begin{bmatrix} \delta P_{x1D}(i) + \delta_x(j) \\ \delta P_{y1D}(j) + a_x(i)\theta_{xy} + \delta_y(i) \end{bmatrix} \quad (1)$$

Where, $\delta p_x(i, j)$ and $\delta p_y(i, j)$ are x direction error and y direction error according to position of x and y, respectively; $\delta p_{x1D}(i)$ is 1D position of X axis and ; $\delta_x(j)$ and $\delta_y(j)$ represent horizontal motion errors at machining point in the X and Y directions; θ_{xy} is perpendicular error between X and Y and a_x is moving distance from origin of X axis.

The position and motion errors in equation (1) was measured by laser interferometer and used as inputs. Figure 2 shows the estimated 2D position errors. The maximum deviations of 2D position errors are 11.59 μm for x-axis and 12.33 μm for y-axis, respectively. The estimated position errors are compared with diagonal

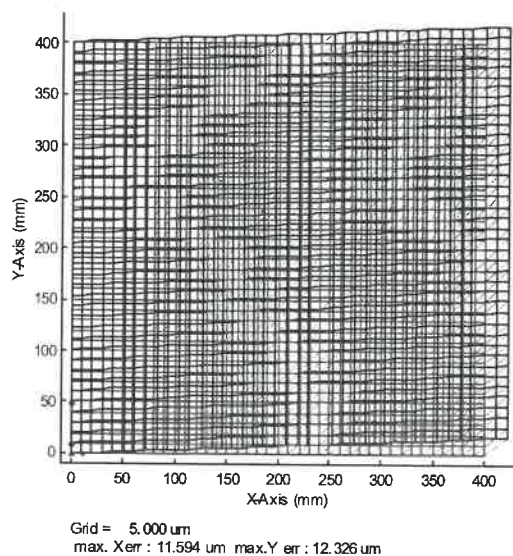


FIGURE 2. Estimated 2D position errors on the XY plane

measurement shown in figure 3. As shown in the figure 3 a steering mirror is located in front of the beam splitter and it turns angle of beams to 45 ° and the reflector is fixed on the corner of the tables. The height of the beam path is maintained at 120 mm, which is average height of the machining point. The measured results are analyzed by ISO 230-6 standard and shown in figure 4. The estimated error laser interferometer measured data along two diagonals, (400,400) to (0,0) and (0,400) to (400,0) are compared with in the figure 4. As shown in the figure the estimated errors are similar in shape and magnitude. It shows that the error model of the equation (1) is efficient to estimate 2D models.

The 1D position error was measured in center of the table and compensated each X and Y axis.

The perpendicular error (θ_{12}) affects the position error $\delta p_y(i, j)$ as indicated by equation (1). It is possible to measure the perpendicular error

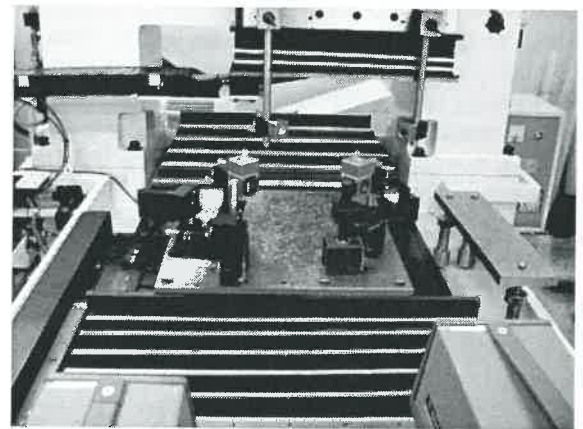


FIGURE 3. Diagonal measurement in the XY plane

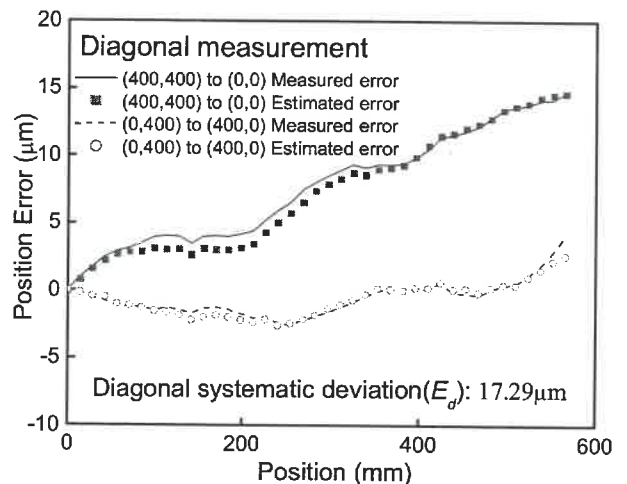


FIGURE 4. Diagonal measurement results before compensation

using two different methods. The easier one is calculating the error directly from the diagonal measurements using equation (2),

$$\theta_{12} = \cos^{-1} \frac{P_d^2 - P_x^2 - P_y^2}{2P_x P_y} \quad (2)$$

where P_d is the measured position along the diagonal direction, P_x is the measured position along the X-axis, and P_y is the measured position along the Y-axis. This is easy to measure and requires only one step of the diagonal measurement process without changing the interferometer system optics.

To compensate 2D errors in XY plane, each errors were corrected by re-shaped command signal. The horizontal motion error was

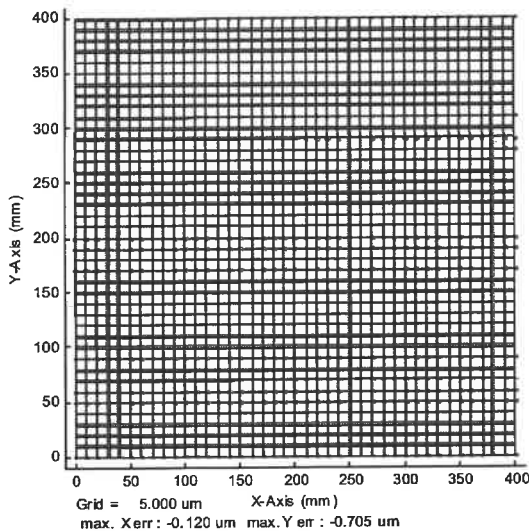


FIGURE 5. Estimated 2D position errors after compensation

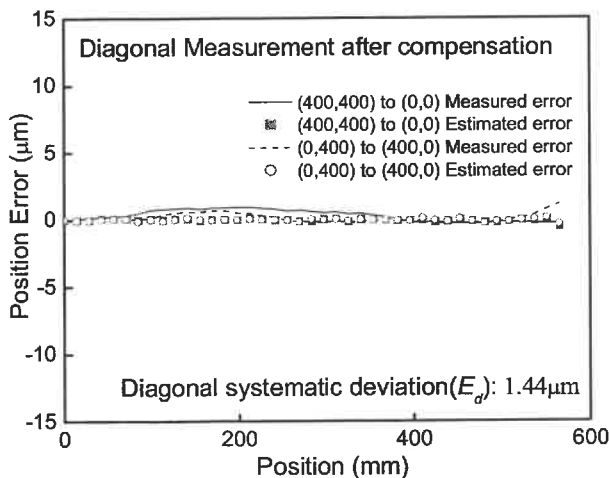


FIGURE 6. Diagonal measurement results after compensation

eliminated by small shift of the cross-axis.

Figure 5 shows the estimated 2D position errors when all the errors describing in equation (1) are compensated. The estimated maximum errors are 0.12 μm and 0.71 μm in X-axis and Y-axis, respectively.

The diagonal measurement errors are shown in figure 6. As shown in figure, the diagonal systematic errors are decreased from 17.3 μm to 1.44 μm from the effect of compensation. The comparison between estimated and measured errors are shown in figure 6. The motion errors, those are used as input of the estimation, were measured in center position of the table but, the diagonal errors were measured in the corner of the table as shown in figure 3. Those could be the main reason of difference between the estimated and measured data. The amount and shapes of difference shown in both figures 4 and 6 are very similar.

ERRORS OF THE ROTATIONAL AXIS

One of main application of the micro patterns on the large surface is making pyramid patterns. To make the pyramid, the triangular prism pattern is grooved firstly along X axis then patterned material is rotated 90° by C axis and same triangular prism is grooved cross of the previous pattern.

During 90° rotating, the height change in the surface are occurred owing to perpendicular errors between C, X and Y axis.

To measure the angle between linear and rotational table, circular interpolation was

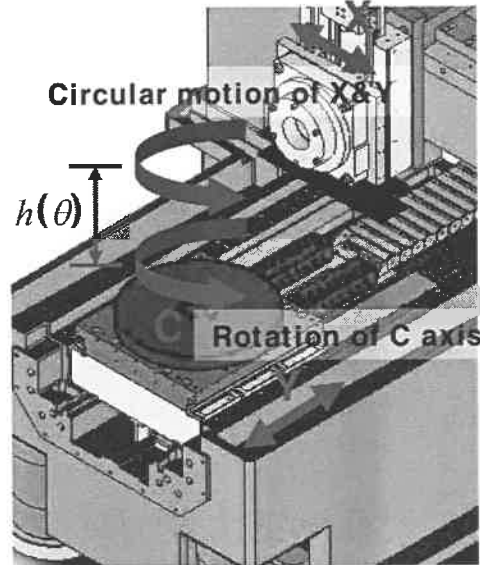


FIGURE 7. Measuring perpendicular errors between C and X-Y axis

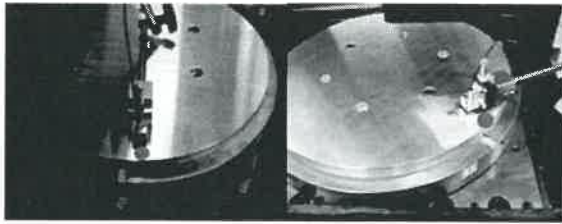


FIGURE 8. Simultaneous three axis motion

simultaneously moved in X and Y axis with rotational table. Which made the measuring target pointing is always same, as shown in figure 8. From the motion the profiles error of the artifact on the C-axis could be eliminated.

The solid line in figure 9(a) is measured data of the deviation of the height ($h(\theta)$), which is main error source of the motion of the tables. To divide the error sources of the tables, tilt motion error of the rotary table, which is shown in dashed line of figure 9(a), was eliminated and the result is shown in solid line of (b), which is supposed to be perpendicular errors of the X and Y axis. And, Perpendicular errors are estimated as dashed line of (b). After eliminating the perpendicular errors, remained errors are supposed to be superposed errors of X and Y tables motion errors, which is shown in (c).

CONCLUSIONS

This paper describes a method to measure and compensate a 4 axes ultra-precision machine

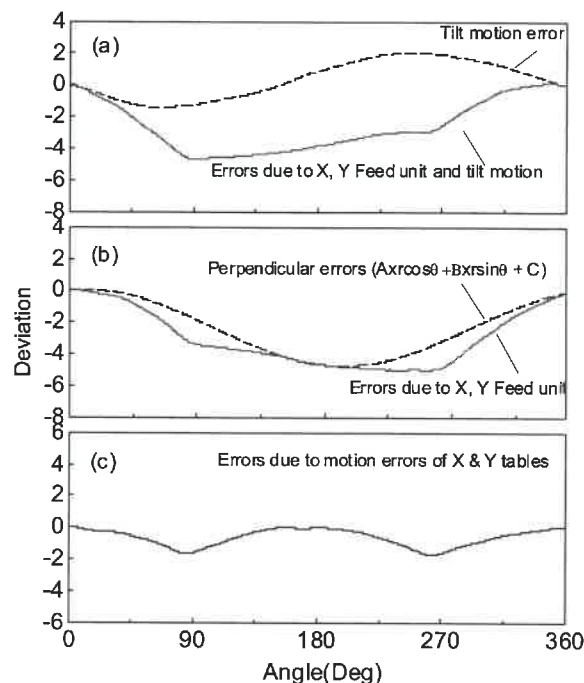


FIGURE 9. Analysis of the Errors between rotational table and linear tables

tool that generates micro patterns on the large surfaces. To compensate a two dimensional positioning error, straightness of two axes in out of plane, squareness between the each axis are important. Positioning errors, straightness and squareness were measured by laser interferometer system. One of difficult problem to measure the error motions is squareness or parallelism of axis between the rotational table and linear axis. It was investigated by suggested simultaneous motion of the three axes.

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A NEW METHOD FOR CHARACTERIZING SPINDLE RADIAL ERROR MOTION: TWO- DIMENSIONAL POINT OF VIEW

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INTRODUCTION

A spindle is often the heart of a machine tool or a measurement instrument. A major criterion to quantify spindle performance is spindle error motion, which is defined as the motion deviation of a spindle under test from an ideal spindle. Standards have been established for specifying and testing spindle radial error motion [1], based on the rotating-sensitive-direction method and the fixed-sensitive-direction method. These methods have been widely used in industry to evaluate spindle performance in applications with only one sensitive direction such as turning and boring of round surfaces. Today, spindles are not restricted to generating axis-symmetric patterns. These processes, such as FTS-assisted turning, machining of non-round holes, and rotary beam writing, are simultaneously sensitive to radial error motion in two dimensions. In this paper, a new two-dimensional (2D) method is presented to specify spindle radial error motion. One result from this 2D analysis is the confirmation of the fundamental (one cycle per revolution or 1 cpr) radial error motion, which was inadvertently removed from motion measurement by circle fitting [1] or Fourier expansion [2]. The experimental measurement results verify that such fundamental radial error motion not only exists, but also is often much larger than all other error motion components.

2D METHOD

At a specified axial location, the spindle radial error motion can be derived from the motion measurement of a marking point. Fig. 1(a) shows a typical spindle metrology setup: displacement measurement probes installed on the metrology frame and a test ball installed on the spindle rotor to mark the ball center point P for motion measurement. This setup allows the ball center radial motion to be measured relative to the metrology frame (OXY). As shown in Fig. 1(b), after removing the ball roundness from the probe measurements, the point P motion in X-Y plane can be represented by a rotation-dependent complex variable

$$v_p(\theta) = x_p(\theta) + jy_p(\theta) \quad (1)$$

where θ is the spindle angular position, and $x_p(\theta)$ and $y_p(\theta)$ are the point P motion components along X and Y axis respectively. In a procedure similar to current standards [1], the test point motion $v_p(\theta)$ can be decomposed into synchronous and asynchronous motion. The synchronous motion,

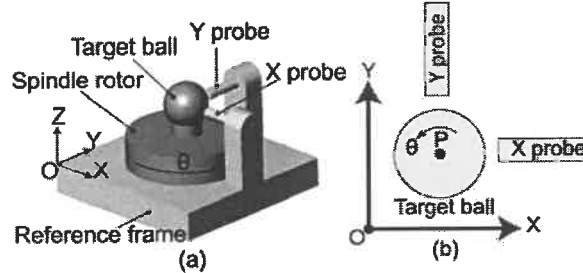


FIGURE 1. Typical metrology setup

$\bar{v}_p(\theta)$, is the synchronized average value of $v_p(\theta)$ over a number of revolutions. As a complex-valued periodic function of θ , $\bar{v}_p(\theta)$ can now be represented as the sum of vectors spinning at harmonic rotation frequencies in both the positive and negative directions:

$$\bar{v}_p(\theta) = \sum_{k=-\infty}^{\infty} \bar{V}_p(k) e^{jk\theta}, \quad (2)$$

where k is any integer number (positive or negative), and $\bar{V}_p(k)$'s k -th Fourier coefficient is

$$\bar{V}_p(k) = \frac{1}{2\pi} \int_0^{2\pi} \bar{v}_p(\theta) e^{-jk\theta} d\theta. \quad (3)$$

Unlike a real-valued function, for the complex-valued function $\bar{v}_p(\theta)$, its Fourier coefficients at positive and negative frequencies are two independent variables with distinct physical meanings. $\bar{V}_p(+1)$ represents the offset of the point P from the spindle rotation center. $\bar{V}_p(0)$ is determined by the DC offset of the probe installation. By eliminating the $k = 0$ and $k = 1$ components from the point P's motion, the spindle error motion (or the spindle rotation center motion) can be derived as:

$$\bar{v}_c(\theta) = \bar{v}_p(\theta) - \bar{V}_p(0) - \bar{V}_p(1)e^{j\theta} = \sum_{\substack{k=-\infty \\ k \neq 0,1}}^{\infty} \bar{V}_p(k) e^{jk\theta}. \quad (4)$$

This result is independent of the location of the test point selection or the target ball installation. In practice, the test ball is always installed as close to the spindle center as possible in order to make the ball center motion measurement easier.

FUNDAMENTAL RADIAL ERROR MOTION

The fundamental (1 cpr) radial error motion has not been recognized in prior art. The presented 2D method is applied to hypothetical examples to illustrate the 1-cpr radial error motion.

Fig. 2 (a) shows the trajectories of points on a perfect spindle rotor as it goes through one full revolution. For a particular point A_1 , its X and Y motions are sinusoidal with identical amplitude, and the Y motion lags the X motion by 90 [deg]. If selecting another point such as A_2 or A_3 , the trajectory is always a concentric circle. By applying equation (4) as shown in Table 1, the error motion of the spindle is always zero, regardless of test point selection. Therefore, spindle A is a perfect spindle.

Fig. 2 (b) shows the trajectories of a spindle with 1 cpr radial error motion as it goes through one full revolution. For a particular point B_1 , the trajectory is also a full circle: both X and Y motions are once-per-revolution sinusoidal with equal amplitude. If the circle fitting method is used for X and Y motion individually to eliminate the 1-cpr components, this spindle would be considered perfect according to [1]. However, Y motion leads X motion by 90 [deg]. The rotor axis of rotation passes through point B_1 . During the spindle rotation, the axis of rotation follows a circular path opposite to the spindle rotation, thus the trajectories of other points on the same spindle rotor could be very different. For example, B_2 's trajectory is an ellipse and B_3 is moving along a straight line back and forth. This spindle is obviously not perfect, despite the fact that X and Y motion measurements of any point on the rotor are 1 cpr. Applying equation (4), the spindle exhibits a fundamental radial error motion rotating opposite to the spindle's main rotation, regardless of which point is selected as shown in Table 1.

Comparing the trajectories in figure 2, we can conclude that the spindle error motion information cannot be found purely based on the X-Y plot. For example A_1 and B_1 both have a full circle motion path but belong to different spindles. Another example is B_1 and B_2 , which both belong to same spindle but have different trajectories in X-Y plot. Using the new 2D method, the 1 cpr error motion of the spindle can be derived regardless of the measured point. This motion is always

represented by $k = -1$ harmonic component of the measured motion path as shown in Table 1.

TABLE 1. Spindle error motion calculation

	Test point motion	Error motion
Spindle A	$\bar{v}_{A_1} = ae^{j\theta} + 0 \cdot e^{-j\theta}$	$\bar{v}_c(\theta) = 0$
	$\bar{v}_{A_2} = \frac{a}{2}e^{j\theta} + 0 \cdot e^{-j\theta}$	
	$\bar{v}_{A_3} = 2ae^{j\theta} + 0 \cdot e^{-j\theta}$	
Spindle B	$\bar{v}_{B_1} = 0 \cdot e^{j\theta} + ae^{-j\theta}$	$\bar{v}_c(\theta) = ae^{-j\theta}$
	$\bar{v}_{B_2} = -\frac{a}{2}e^{j\theta} + ae^{-j\theta}$	
	$\bar{v}_{B_3} = ae^{j\theta} + ae^{-j\theta}$	

EXPERIMENT

The 2D method is used to analyze radial error motion of two spindles: a rolling-element bearing spindle in a Mori Seiki high-speed 5 Axis machine center, and an aerostatic bearing spindle (10R Block-Head by Professional Instruments). Here the results show that 1 cpr radial error motion exists in both cases and is the dominant error component.

BALL BEARING SPINDLE EXPERIMENT

Fig. 3 shows the experimental setup used to measure the spindle error motion. Two 10- μ m stroke probes (C5-D by Lion Precision), clamped on the machine table, measure the motion of a target ball center (20-nm roundness) in X-Y plane.

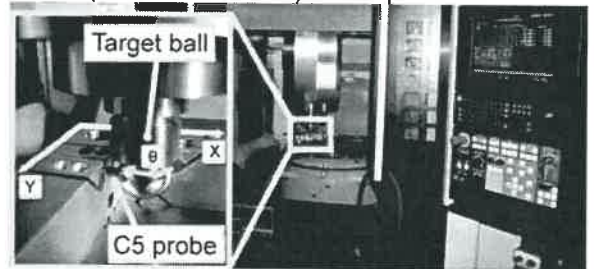


FIGURE 3. Experimental setup

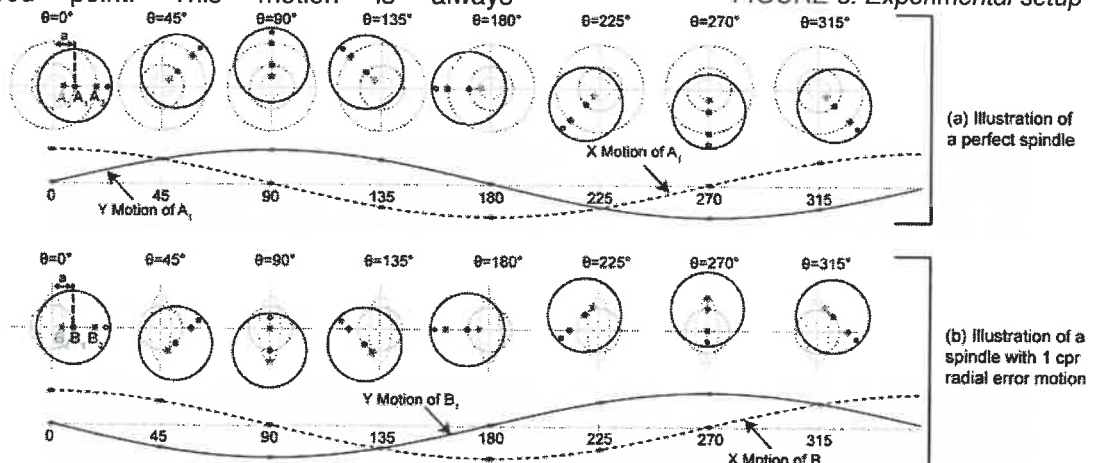


FIGURE 2. Perfect spindle vs. spindle with 1 cpr radial error motion.

Fig. 4. shows the test ball synchronous motion measured by X and Y probes at 4000 rpm along with their 1 cpr components in dash lines. Although the test ball roundness was not removed, the 20nm round ball can be treated as a perfect one for a spindle with μm motion. The 1cpr motion amplitudes in X and Y directions are very different; therefore, the test ball center is following an elliptical path in the X-Y plane, with the ball center offset $\bar{V}_p(+1) = 2.92 \mu\text{m}$ and 1 cpr spindle error motion $\bar{V}_p(-1) = 1.50 \mu\text{m}$.

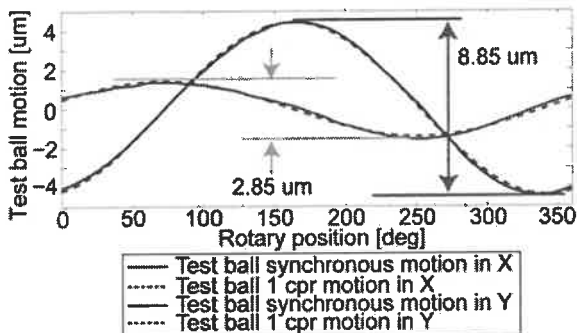


FIGURE 4. Synchronous Motion at 4000 rpm

Fig. 5 compares three polar plots at 4000 rpm: the ball center synchronous motion measured by the X probe; the synchronous error motion along X specified in [1] by completely removing the 1 cpr motion component of the X probe measurement; and the spindle synchronous error motion along X using the 2D method. It is shown that the 1 cpr spindle radial motion is neglected in the current standard.

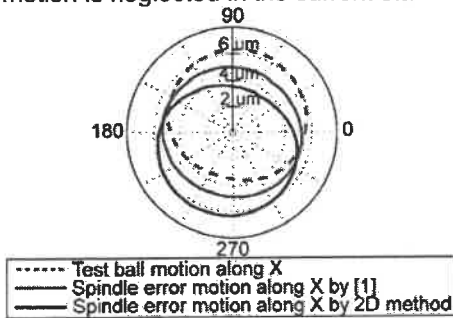


FIGURE 5. Polar plots of synchronous motion along X-direction at 4000 rpm.

A major cause of the spindle 1 cpr radial error motion is the interaction between the unbalanced rotor and the asymmetric structural stiffness of the spindle along X and Y. For most machines, the structural stiffness in X and Y are different. Fig. 6 shows the results of a hammer test on the spindle rotor. The results indicate that the structural stiffness plots along X and Y directions are different at most frequencies. When the spindle rotor is excited by its unbalanced mass, the motion along X and Y will be unequal, due to the different stiffness in each direction. Consequently, the spindle will exhibit 1 cpr radial error motion.

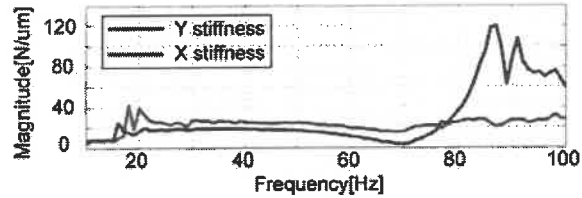


FIGURE 6. Spindle stiffness in X and Y directions.

Fig. 7. shows the magnitude and phase of $\bar{V}_p(+1)$ and $\bar{V}_p(-1)$ of the measured point for a range of speeds. The radial motion becomes larger at 4000 rpm, due to structural resonance at 70 Hz along the Y direction.

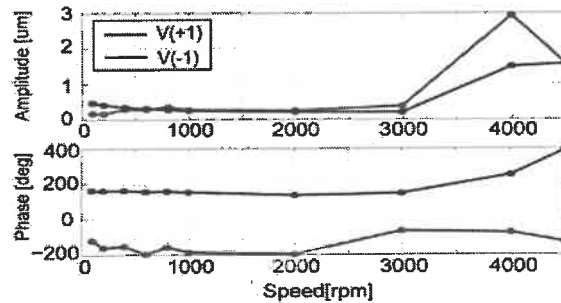


FIGURE 7. Fourier coefficients across speed.

AIR BEARING SPINDLE EXPERIMENT

Fig. 8 shows the experimental setups used to measure the radial error motion of a precision air bearing spindle. Three independent setups are used to measure the error motion of the spindle: (a) a 50- μm stroke probe (C2-A by Lion Precision) to measure the motion of a target ball (20-nm round) sequentially from two orthogonal directions; (b) two 10- μm probes (C5-D by Lion Precision) to measure the center motion of an encoder ring (200-mm diameter) in two orthogonal directions, at the same axial location of the target ball; (c) four encoder heads equally spaced around the encoder ring to measure the spindle rotation and the ring center radial motion in two orthogonal directions.

Fig. 9 shows the synchronous motion of the test ball center using the setup (a), at 500 rpm. The ball roundness is also included in the results, but the roundness contains only 2nd and higher order components and thus has no influence on the 1 cpr spindle error motion measurement. The 1cpr motion amplitude along X is 142 nm, while along Y it is 12 nm. As a result, the spindle has a fundamental radial error motion of 68 nm, spinning opposite to the spindle rotation. This number is much larger than the 25 nm radial error motion specified by the ANSI method [1] for this Blockhead spindle.

Fig. 10 shows the measurement results from three setups. For both setups (a) and (b), the spindle radial error motion is measured at various constant speeds and 500 revolutions are used to calculate the synchronous motion. For setup (c), the measurement is during the spindle freely slowing

down and one revolution data at each speed is used to calculate 1 cpr error motion. All results show that the 1 cpr error motion amplitude changes by a few nanometers across the measured speed range. The results from (b) and (c) agree well. The difference between (a) and (b) results is caused by the ball target radius, which makes the probe reading in (a) appear to be a bit more than the actual motion.

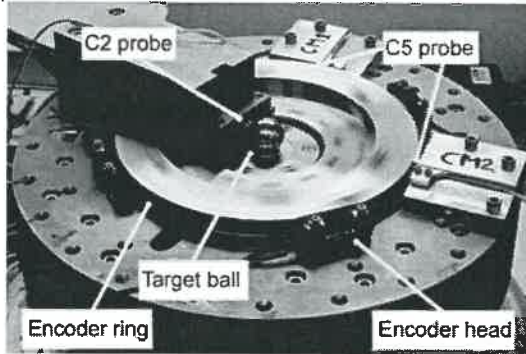


FIGURE 8. Experimental setup.

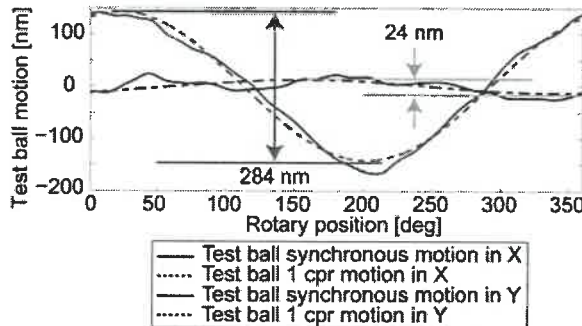


FIGURE 9. Synchronous motion at 500 rpm

EFFECT of ERROR MOTION on APPLICATIONS

For spindle applications sensitive to error motion along only one particular radial direction, the spindle radial error motion along this specified direction is

$$\rho(\theta) = \text{Re}[\bar{v}_c(\theta)e^{-j\phi}], \quad (5)$$

where ϕ is the angle between the specified sensitive direction and the X axis as shown in Fig. 11. If an application is sensitive to single fixed direction, then ϕ is a constant number and $\bar{v}_p(-1)$ will have only second-order effect on such application. If an application is sensitive to error motion along single radial direction that rotates with the rotor, then $\phi = \theta$ and the error motion along the specified direction is

$$\rho(\theta) = \text{Re}[\bar{v}_c(\theta)e^{-j\theta}] = \text{Re}\left[\sum_{\substack{k=-\infty \\ k \neq 0,1}}^{k=\infty} \bar{v}_p(k)e^{j(k-1)\theta}\right] \cdot (6)$$

Due to the modulation effect in (6), the error motion $\bar{v}_p(-1)$ will cause a 2 cpr component in $\rho(\theta)$; the error motion $\bar{v}_p(2)$ will result in a 1 cpr

component in $\rho(\theta)$, but bring only second-order affect on the application. All these agree with the error motions specified in [1], which is not the error motion along the specified direction, but the error motion components to which the particular types of applications are sensitive. For applications with two radial sensitive directions, the full error motion from (4) should be used.

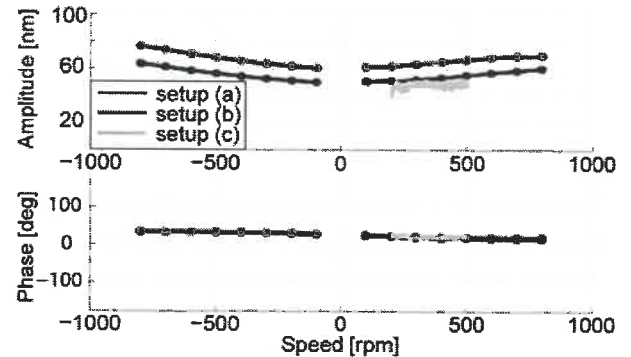


FIGURE 10. Fundamental radial error motion.

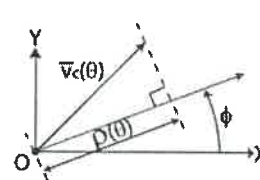


FIGURE 11: error motion along a specified sensitive direction (the red line).

CONCLUSION

Theories and experimental results of a new 2D method are presented to evaluate spindle radial error motion. Advantages of the 2D method include: error motion description in 2D; error motion description along any specified single radial direction; prediction of errors caused by spindle radial error motions in all spindle applications. In contrast, the method in [1] is limited to specify errors caused by spindle radial error motion in applications that are sensitive to only single radial direction.

ACKNOWLEDGEMENT

Darcy Montgomery of Kodak first questioned about ANSI B89.3.4 [1] and pointed out that in [1] the fundamental component was mistakenly removed from probe measurements. His questions and comments motivated the authors to develop the presented 2D method for a more rigorous treatment of the spindle radial error motion. KLA-Tencor provided most of the equipments. Darya Amin-Shahidi, a former graduate at UBC, designed the air-bearing test setup. MAL lab of UBC provided the access to the Mori Seiki machine.

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CONCURRENT MEASUREMENT METHOD FOR SPINDLE RADIAL, AXIAL AND ANGULAR MOTIONS USING CONCENTRIC GRATING INTERFEROMETER

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INTRODUCTION

Ideal spindles as basic mechanical components must have only one-dimensional rotary motion degree of freedom; therefore, extra five-dimensional motion degrees of freedom (= radial (2 degrees) + axial (1 degree) + angular (2 degrees)) are identified as motion errors, which must be eliminated or reduced. Since ultra precision fabrication machines and information technology instruments such as hard disks, have progressed very rapidly, reductions in spindle rotating errors (radial, axial and angular motions) to nanometer and sub- μ radian levels in such industries are required. In this paper, we propose a novel method for the concurrent measurement of five degree motion errors (= radial, axial and angular motions) using a concentric circle grating and laser interferometers, and discuss its application to ultra precise and high-speed spindles. In this method, a concentric circle grating plate with a fine pitch is installed on top of the spindle of interest and is used as the reference artifact. Three optical sensors, which consist of two interferometers, are fixed over the grating. One interferometer detects an interference signal between reflection lights from a fixed mirror and the grating for vertical displacement measurement. The other interferometer detects an interference signal between ± 1 st diffraction lights from the grating for lateral displacement measurement. Using three optical sensors, radial, axial and angular motions of the spindle, can be measured concurrently. Because the concentric circle grating plate is not voluminous and not heavy, this method is effective for any spindle, and does not affect the spindle original rotational motion. Since this method is based on wide-bandwidth photo sensors, it is possible to apply it to high-

rotational-speed spindles. Moreover, this method is suitable for maintaining the traceability against the meter definition because it uses laser interferometers. The use of the concentric circle grating has been proposed to measure spindle radial motion error by the other researchers [1][2]. In this paper, we discuss the measurement principle, preliminary experiment, conclusion and future study.

MEASUREMENT PRINCIPLE

Figure 1 shows a schematic diagram of the two-dimensional displacement measuring interferometer. In the figure, the X-, Y- and Z-axes indicate the grating pitch, the ruling and height directions, respectively; therefore, the XY plane is parallel to the grating plane. The two-dimensional displacement measurement is performed by X- and Z-axes displacement measuring interferometers, as shown in Figure 1.

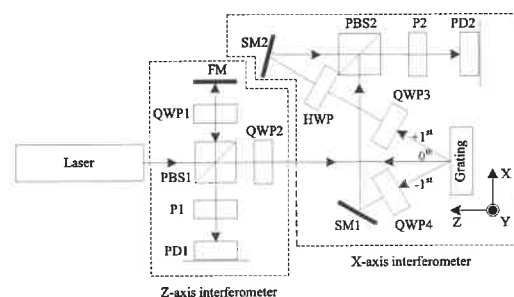


FIGURE 1. Schematic diagram of two-dimensional displacement measuring interferometer; PBS: polarised beam splitter, QWP: quarter-wave plate, HWP: half-wave plate, FM: fixed mirror, SM: steering mirror, PD: photodiode and P: polariser.

A light beam emitted from the laser is incident on polarised beam splitter 1 (PBS1), and then, it is divided into two beams. The first beam reflected from PBS1 enters the fixed mirror (FM) via quarter-wave plate 1 (QWP1) and bounces from FM and reaches photodiode 1 (PD1) via PBS1 and polarizer 1 (P1). The second beam, which goes through PBS1, reaches the one-dimensional grating via QWP2. In Figure 1, we assume that the incident angle θ_i of the second beam to the grating is almost null. Thus, the reflected light (= 0th-order diffraction light) reaches PD1 via QWP2, PBS1 and P1. At PD1, it is possible to detect an interferometer fringe between the reflection lights from FM and the grating, and to measure the displacement along the Z-axis of the grating. The incident light on the grating can produce ± 1 st-order diffraction lights. The diffraction light direction is derived using the vector format [3] from

$$S' \times \varepsilon - S \times \varepsilon = m \frac{\lambda}{nd} q, \quad (1)$$

where S , S' , ε , q , m , d , n and λ represent a unit vector along the incident light direction, a unit vector along the diffraction light direction, a unit vector normal to the grating plane, a unit vector along the ruling direction, a diffraction order, a grating pitch, a refractive index and a vacuum wavelength, respectively.

In the figure, the ε and q vectors are the same as the unit vectors along the Z- and Y-axes. If the incident angle on the XZ plane is θ_{iXZ} ($\theta_{iXZ} \ll 1$), the ± 1 st-order diffraction angles θ_{+1} and θ_{-1} (on XZ plane) can be derived from

$$d(\sin \theta_{+1} - \sin \theta_{iXZ}) = +\lambda \quad (2)$$

$$d(\sin \theta_{-1} - \sin \theta_{iXZ}) = -\lambda \quad (3)$$

We assume that $n = 1$.

In Figure 1, the interference fringe between ± 1 st-order diffraction lights is formed on PD2 via QWP 3/4, half-wave plate (HWP), steering mirror 1/2 (SM1/2), PBS2 and P2. We assume that Δx , Δy and Δz are the displacement shifts of the grating along the X-, Y- and Z-axes, respectively. We also assume that the small incident angle θ_{iYZ} ($\theta_{iYZ} \ll 1$) on the YZ plane

and the small incident angle θ_{iZ} ($\theta_{iZ} \ll 1$) between the incident light and the Z-axis affect the interference fringe. In figure 1, the output signals (I_{PD1} and I_{PD2}) from PD1 and PD2 can be expressed as following format

$$I_{PD1} = A \left\{ 1 + \Gamma_A \cos \left(\frac{4\pi z}{\lambda} \right) \right\} \quad (4)$$

$$I_{PD2} = B \left\{ 1 + \Gamma_B \cos \left(\frac{4\pi x}{d} \right) \right\} \quad (5)$$

where A (or B), z (or x), λ , d and Γ_A (or Γ_B) are signal amplitude, Z (or X) axis displacement of the grating, a wavelength of illuminated light and a grating pitch and a fringe contrast, respectively.

Because the incident angle to the grating is approximately 0 degree, we can obtain equations (4) and (5). By the combination of the two interferometers, the optical sensor can measure lateral (X) and vertical (Z) displacements of the concentric grating simultaneously.

Figure 2 shows the measurement principle of the spindle motion error using the two-dimensional displacements measuring interferometers.

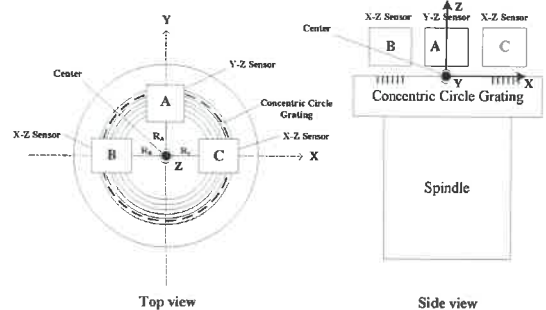


FIGURE 2. Measurement principle for radial, axial and angular motions using two-dimensional displacement measuring interferometer. A, B and C are two-dimensional displacement interferometers, A: YZ sensor, B and C: XZ sensor.

A concentric circle grating is set on top of the spindle of interest. Three two-dimensional displacement measuring interferometers A, B and C are fixed over the concentric circle grating. The spindle rotation center must be nearly

aligned to the center of the concentric grating within 1 μm or less. The grating plane must also be aligned to be parallel with the XY plane within 10arcsec. In the Figure 2, the center is selected as the origin point of the XYZ coordinate system. The two-dimensional displacement measuring interferometer A is located along the Y-axis with the distance R_A from the center and measures the Y-axis and Z-axis displacements y_A and z_A using the concentric circle grating, whereas the interferometers B and C are located along the X-axis with the distances R_B and R_C from the center, and measure the X-axis and Z-axis displacements x_B , z_B , x_C and z_C using the concentric circle grating. From the measured displacements by the three interferometers, the radial R_x , R_y , axial R_z and angular θ_x , θ_y motions can be derived from

$$R_x = \frac{1}{2}(x_B + x_C) \quad (6),$$

$$R_y = y_A \quad (7),$$

$$R_z = \frac{1}{3}(z_A + z_B + z_C) \quad (8),$$

$$\theta_x = \frac{1}{R} \left\{ z_A - \frac{(z_B + z_C)}{2} \right\} \quad (9),$$

$$\theta_y = \frac{2}{R} (z_B - z_C) \quad (10),$$

$$R = \frac{1}{3}(R_A + R_B + R_C) \quad (11),$$

where R is the averaged distance of the interferometers A, B and C from the center.

In general, the grating must exhibit artefact errors (= roundness, flatness and irregular pitch); however, it is possible to apply the error separation methods (multistep, multipoint) to the concentric circle grating. Therefore, another arrangement of interferometers A, B and C is possible.

PRELIMINARY EXPERIMENT

In order to confirm the equation (4) and (5), we constructed the grating interferometer using a one-dimensional grating as shown in Figure 3. In the experiment, the grating was set on the XZ-axes stage and moved with a piezoelectric

(PZT) actuators along both the X- and Z-axes. Instead of the concentric circle grating, a one-dimensional grating with 1200 lines/mm was utilized. A He-Ne laser ($\lambda = 633\text{nm}$) was used as the light source. In our experiment, interference signals on PD1 and PD2 were compared with the driving signal of PZT actuators.

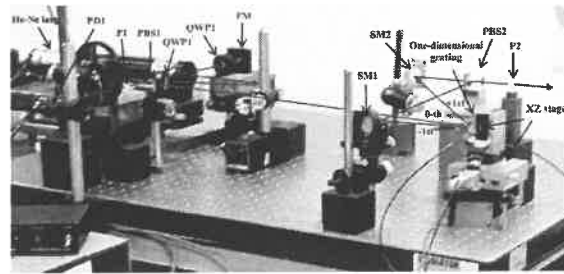


FIGURE 3. Schematic photograph of experimental system.

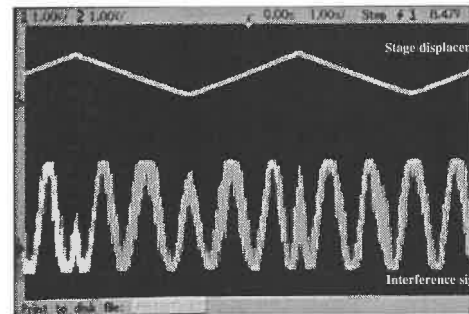


FIGURE 4. Results z displacement signals from the photo detector, PD1.

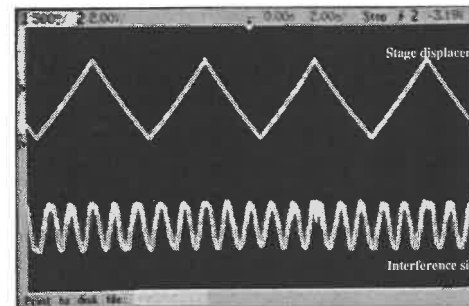


FIGURE 5. Results x displacement signals from the photo detector, PD2.

Figures 4 and 5 show the comparison results. Figure 4 shows the time variations of the X-axis driving signal and the interference fringe of ± 1 st-order diffraction lights obtained when the Z-axis movement was stopped. Figure 5 shows the time variation of the Z-axis driving signal and

the interference fringe of the reflected lights from FM and the grating obtained when the X-axis movement was stopped.

In Figure 4, the applied displacement amplitude to the X-axis stage is $1.3\text{ }\mu\text{m}$, and the observed fringe period of the interference is about 4. This shows that the basic period is $0.4\text{ }\mu\text{m}$, and $d/2$ is 417nm . In Figure 5, the applied displacement amplitude to the Z-axis stage is $1.1\text{ }\mu\text{m}$, the observed period of the interference signal is nearly $0.3\text{ }\mu\text{m}$, and $\lambda/2$ is 317nm . The experiments results show that the proposed method enables the measurement of the X-axis and Z-axis displacements from the interference fringes.

CONCLUSION

In this paper, to measure the spindle radial, axial and angular motions simultaneously, we propose a novel method using a concentric circle grating and interferometers. The measurement principle of a two-dimensional displacement measuring interferometer using a one-dimensional grating is clarified. The error analysis of the interferometer is performed from the views of radial, axial and angular motion measurements. For a small motion error, the interferometer can be used to measure the lateral and vertical axes displacements with high speed.

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SYNCHRONOUS MOTION OF SPINDLES

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INTRODUCTION

Motion of a perfect spindle is rotation about an axis with no radial, axial, or tilt motion involved [1][2]. Such perfect motion is theoretical only, and these other components always exist to a greater or lesser extent. This report is primarily concerned with radial motion, and more specifically synchronous radial motion; synchronous motion occurs at frequencies that are integer multiples of the rotation frequency [1][2]. Following an analysis of synchronous radial motion, there are brief discussions of error motion compensation, residual motion, artifact centering, tilt motion, and error motion prediction. Test results are also presented.

SYNCHRONOUS MOTION

As noted, motion of a perfect spindle involves only rotation about a stationary axis. For a perfect spindle z indicates the axis of rotation and θ is the orientation of the spindle relative to the x -axis. This motion can be defined by:

$$\begin{aligned}x &= y = z = \alpha_x = \alpha_y = 0 \\ \alpha_z &= \theta\end{aligned}$$

For a real spindle, none of x , y , z , α_x , or α_y , are zero. Synchronous spindle motion in each of these five degrees of freedom can be defined as independent functions of θ . Radial motion occurs in the x and y directions, and synchronous radial motion specifically can be defined as:

$$\begin{aligned}x &= \sum_{j=0}^{\infty} [a_{jx} \cos(j\theta) + b_{jx} \sin(j\theta)] \\ y &= \sum_{j=0}^{\infty} [a_{jy} \cos(j\theta) + b_{jy} \sin(j\theta)]\end{aligned}$$

where, j are integers, and a , and b are constants. $j=0$ corresponds to an offset of the axis of rotation with respect to the reference coordinate system. $j=1,2,3$, etc. correspond to first, second, third, etc. harmonic motions of the spindle with respect to the fixed reference coordinate system. Asynchronous motion occurs at non-integer multiples of the fundamental frequency [1][2] and although it is of great practical importance, it is assumed to be

zero for the remainder of this report. This assumption is valid because in general, asynchronous and synchronous motion can be separated mathematically [1][2], which allows us to consider both types of spindle motion independently.

In order to measure the error motion of a real spindle, a precision artifact (commonly a nearly perfect sphere) is mounted to the spindle, and the motion of the artifact is measured with respect to one or two stationary indicators. The artifact is assumed here to have perfect form, although there are well known methods to separate artifact form error from the measured artifact motion [1][2]. Of course, the artifact will not be located exactly on the z axis, and for a perfect spindle the motion of the artifact can be defined by:

$$\begin{aligned}x_a &= A_x \cos\theta - A_y \sin\theta \\ y_a &= A_y \cos\theta + A_x \sin\theta\end{aligned}$$

where A_x and A_y are the distance of the artifact from the z axis in the x and y directions respectively.

Measurement of the artifact is typically made with mechanical, optical, or electrical indicators [1][2]. All of these can be located at some arbitrary position, and zeroed at another arbitrary position. Additionally, the artifact size will have a direct influence on the static part of the measurement. The measurement will have an offset resulting from these factors which can be defined by:

$$\begin{aligned}x_s &= B_{sx} \\ y_s &= B_{sy}\end{aligned}$$

When the synchronous radial motion of the spindle is combined with the motion of the eccentric artifact and the measurement offset term, the measured result can be defined by:

$$\begin{aligned}X &= \sum_{j=0}^{\infty} [a_{jx} \cos(j\theta) + b_{jx} \sin(j\theta)] \\ &\quad + A_x \cos\theta - A_y \sin\theta + B_{sx}\end{aligned}$$

$$Y = \sum_{j=0}^{\infty} [a_{jy} \cos(j\theta) + b_{jy} \sin(j\theta)] \\ + A_y \cos\theta + A_x \sin\theta + B_{sy}$$

This is what would be measured with two perpendicular radial fixed indicators. Values increase as the ball moves closer to the indicator. The zero harmonic measurement includes spindle motion terms a_{0x} and a_{0y} , and the measurement offset values. The first harmonic is a combination of spindle motion and artifact eccentricity which includes the terms a_{1x} , b_{1x} , a_{1y} , b_{1y} , A_x , and A_y . Although it is standard practice to set the fundamental term to zero [1][2], we have taken the approach that applications exist where the corresponding spindle motion is relevant. Theoretically, the fundamental radial error motion can be circular, elliptical, or linear. The second and higher order harmonics of X , Y are functions of the corresponding spindle harmonic motions.

Rotating indicator measurements are considered now. In this case the measurement axes X' , Y' are rotating with the spindle, and the precision artifact is stationary. Spindle motion x , y in fixed coordinates can be transformed to rotating coordinates by:

$$x' = x \cos(\theta) + y \sin(\theta) \\ y' = -x \sin(\theta) + y \cos(\theta)$$

If the artifact is at position A_x' , A_y' and the rotating measurement offsets are B_{sx}' and B_{sy}' then the total measured values can be defined by:

$$X' = - \sum_{j=0}^{\infty} [a_{jx} \cos(\theta) \cos(j\theta) - b_{jx} \cos(\theta) \sin(j\theta) \\ - a_{jy} \sin(\theta) \cos(j\theta) - b_{jy} \sin(\theta) \sin(j\theta)] \\ + A_x' \cos(\theta) + A_y' \sin(\theta) + B_{sx}'$$

$$Y' = - \sum_{j=0}^{\infty} [-a_{jx} \sin(\theta) \cos(j\theta) + b_{jx} \sin(\theta) \sin(j\theta) \\ - a_{jy} \cos(\theta) \cos(j\theta) - b_{jy} \cos(\theta) \sin(j\theta)] \\ - A_x' \sin(\theta) + A_y' \cos(\theta) + B_{sy}'$$

The spindle motion terms are negative because the indicators are attached to the rotating spindle.

Recalling three trigonometric identities [3]:

$$\sin\beta \sin\gamma = \frac{1}{2}[\cos(\beta-\gamma) - \cos(\beta+\gamma)] \\ \sin\beta \cos\gamma = \frac{1}{2}[\sin(\beta+\gamma) + \sin(\beta-\gamma)] \\ \cos\beta \cos\gamma = \frac{1}{2}[\cos(\beta-\gamma) + \cos(\beta+\gamma)]$$

and rearranging, the total rotating sensitive measurement can alternatively be defined by:

$$X' = -\frac{1}{2} \sum_{j=0}^{\infty} [(a_{jx}+b_{jy}) \cos(j-1)\theta + (b_{jx}-a_{jy}) \sin(j-1)\theta \\ + (a_{jx}-b_{jy}) \cos(j+1)\theta + (a_{jy}+b_{jx}) \sin(j+1)\theta] \\ + A_x' \cos(\theta) + A_y' \sin(\theta) + B_{sx}'$$

$$Y' = -\frac{1}{2} \sum_{j=0}^{\infty} [(a_{jy}-b_{jx}) \cos(j-1)\theta + (b_{jy}+a_{jx}) \sin(j-1)\theta \\ + (a_{jy}+b_{jx}) \cos(j+1)\theta + (-a_{jx}+b_{jy}) \sin(j+1)\theta] \\ - A_x' \sin(\theta) + A_y' \cos(\theta) + B_{sy}'$$

From these equations it is clear that for rotating indicator measurements, the zero harmonic component of the measurement is the result of fundamental spindle motion and measurement offset. Furthermore, the fundamental component is the result of zero and second harmonic spindle motions and artifact position. If it is assumed that the z axis corresponds with the axis of rotation (zero harmonic spindle motion is equal to zero), then the remaining fundamental component is the result of second harmonic spindle motion and artifact position. The higher harmonic components of the measurement are the result of ± 1 spindle harmonic motions.

Because our spindle is a rigid body, the physical motion is independent of measurement method, measurement offsets, or artifact locations. The spindle motion can alternatively be defined by the location of the instantaneous center of rotation:

$$x_c = x - [dy/dt]/[d\theta/dt] \\ = \sum_{j=0}^{\infty} [(a_{jx} - jb_{jy}) \cos(j\theta) + (ja_{jy} + b_{jx}) \sin(j\theta)] \\ y_c = y + [dx/dt]/[d\theta/dt] \\ = \sum_{j=0}^{\infty} [(a_{jy} + jb_{jx}) \cos(j\theta) - (ja_{jx} - b_{jy}) \sin(j\theta)]$$

It is notable that the contribution of fundamental error motion is dependent on the terms $(a_{1x}-b_{1y})$ and $(a_{1y}+b_{1x})$ only. Therefore, there are an infinite number of solutions for the fundamental error motion constants that will result in the same physical motion of the spindle. In the absence of higher order error motions, the locus of instantaneous centers is a circle with radius $R_c = [(a_{1x} - b_{1y})^2 + (a_{1y} + b_{1x})^2]^{1/2}$, centered at the axis of rotation, traversing a path at the same frequency but in the opposite direction to spindle rotation. This is independent of artifact position.

DISCUSSION

In order to define the magnitude of fundamental spindle error, R_c or $R_c/2$ are adequate. However, if it is necessary to compensate for the fundamental radial error motion, then the following equation can be used:

$$\Delta_y = y_1(\theta) - x_1(\theta - \pi/2) \\ = (a_{1y} + b_{1x}) \cos(\theta) - (a_{1x} - b_{1y}) \sin(\theta)$$

This is the correction required for a tool located on the y axis to create a trace that is concentric to a trace created by an uncorrected tool on the x axis. Similar corrections can be made for higher order harmonics, tangential errors, and for tools at arbitrary angles. No correction is required for fundamental error motion if the tools are spaced 180° apart.

Residual error motion is usually defined as the difference between average and fundamental measurements [1][2]. An alternative definition includes residual fundamental motion of: $a_{1x} = -b_{1y}$ and $a_{1y} = b_{1x}$. This defines the error motion as a circle with radius $R_c/2$, centered at the axis of rotation, traversing a path at the same frequency but in the opposite direction to the spindle rotation. This definition results in no contribution from fundamental motion to zero order measured values when using rotating sensors. Note that the values of A_x , A_y , B_{sx} , and B_{sy} are dependent on how the fundamental error is defined.

It is clear that even if the only error motion is fundamental, that it is impossible to center the artifact so that the measured data does not possess any once per revolution content. Although it is possible to move the artifact to a position where it will not have any once per revolution content in a specific direction, it will still contain once per revolution content in every other direction (except at 180°). For example, if the artifact is moved to position: $A_x = -(a_{1x} - b_{1y})$, and $A_y = (a_{1y} + b_{1x})$ it will have no once per revolution motion in the x direction. An artifact is considered centered when $A_x = A_y = 0$.

Synchronous tilt motion of a spindle (still assumed to be a rigid body) associated with two axially separated radial displacements can be calculated directly from the constants $i a_i$ and $i b_i$:

$$\alpha_x(\theta) = \sum_{j=0}^{\infty} \alpha_{jx}(\theta) = \sum_{j=0}^{\infty} [(2a_{jy} - 1a_{jy})\cos(\theta) + (2b_{jy} - 1b_{jy})\sin(\theta)] / \delta z \\ \alpha_y(\theta) = \sum_{j=0}^{\infty} \alpha_{jy}(\theta) = \sum_{j=0}^{\infty} [(2a_{jx} - 1a_{jx})\cos(\theta) + (2b_{jx} - 1b_{jx})\sin(\theta)] / \delta z$$

where i corresponds to either measurement 1 or 2, and δz is the difference in z location of the two measurements.

From measured spindle data it is quite simple to determine the values of all constants and artifact positions if for example the definition of residual error motion given above is followed. In this case, the constants can be determined by:

$$B_{sx} = X_0 \\ B_{sy} = Y_0$$

$$a_{1x} = -b_{1y} = [X_1(0) - Y_1(\pi/2)]/2 \\ a_{1y} = b_{1x} = [Y_1(0) + X_1(\pi/2)]/2$$

$$A_x = X_1(0) - a_{1x} \\ A_y = Y_1(0) - a_{1y}$$

$$B_{sx}' = X_0' \\ B_{sy}' = Y_0'$$

For $j=2,3,4\dots$

$$a_{jx} = X_j(0) \\ a_{jy} = Y_j(0) \\ b_{jx} = X_j(\pi/2j) \\ b_{jy} = Y_j(\pi/2j)$$

$$A_x' = X_1'(0) + 1/2(a_{2x} + b_{2y}) \\ A_y' = Y_1'(0) + 1/2(a_{2y} - b_{2x})$$

Where $X_j(\phi)$ and $X_j'(\phi)$ are the values of filtered j^{th} harmonic measured X and X' data respectively at angular position $\theta = \phi$.

TEST RESULTS

In order to validate the theory, a test spindle was constructed to have known physical fundamental error motion. This was achieved by placing a Scotch yoke in series with a rotor to provide large (about 20 mm) fundamental error motions, and to allow similarly large artifact eccentricity. The test spindle was equipped with a microswitch triggered LED to indicate an index position, and had a large planar face perpendicular to the axis for recording artifact and sensor positions. Based on the construction of the test spindle, the error motion constants were estimated to be: $a_{1x} = 21$ mm, $b_{1x} = 3$ mm, and all others were zero. From this we can say that $R_c = 21.2$ mm, and $\Delta_y = -21\sin(\theta) + 3\cos(\theta)$. Additionally, the corresponding residual fundamental error motion constants (using above definition) are: $a_{1x} = 10.5$ mm, $b_{1x} = 1.5$ mm, $a_{1y} = 1.5$ mm, and $b_{1y} = -10.5$ mm.

Three tests were conducted:

1. fixed sensors with: $B_{sx} = -42$, $B_{sy} = -17$, $A_x = 3$, $A_y = 1$, and measurement increment

- of $2\pi/32$ Radians. If residual fundamental error motion is defined as above, then $A_x = 13.5$, and $A_y = -0.5$.
- fixed sensors with: $B_{sx} = -42$, $B_{sy} = -17$, $A_x = -25$, $A_y = 14$, and measurement increment of $2\pi/32$ Radians. If residual fundamental error motion is defined as above, then $A_x = -14.5$, and $A_y = 12.5$.
 - rotating sensors with: $B_{sx}' = -45$, $B_{sy}' = -47$, $A_x' = 32$, $A_y' = -26$, and measurement increment of $2\pi/32$ Radians. If residual fundamental error motion is defined as above, then $B_{sx}' = -55.5$, and $B_{sy}' = -48.5$.

Positions of artifacts and sensors were marked on the spindle face with a pen, then measured with a vernier caliper. Total measurement accuracy was estimated to be about ± 2 mm. FFT analysis of the experimental data resulted in the values given in Table 1.

TABLE 1. Calculated values from measured data for tests #1, #2, & #3.

Fixed sensors	Test #1	Test #2	Rotating sensors	Test #3
X_0	-40.7	-41.0	X_0'	-56.6
Y_0	-15.7	-16.0	Y_0'	-46.4
$X_1(0)$	23.5	-4.2	$X_1'(0)$	31.7
$Y_1(0)$	1.5	14.5	$Y_1'(0)$	-26.4
$X_1(\pi/2)$	1.8	-11.2	$X_2'(0)$	-9.8
$Y_1(\pi/2)$	3.5	-23.8	$Y_2'(0)$	-1.5
$X_2(0)$	-0.2	-0.1	$X_2'(\pi/4)$	-1.6
$Y_2(0)$	0.0	0.1	$Y_2'(\pi/4)$	10.1
$X_2(\pi/4)$	0.2	0.2	-	-
$Y_2(\pi/4)$	0.2	0.4	-	-

Figure 1 shows error motion results plotted three ways for each test: 1) measured data, 2) calculated data based on constants estimated from measured data alone, and 3) calculated data based on known physical construction of the test spindle. The calculated data based on constants estimated from measured data alone include zero, first, and second harmonic terms.

Calculated values of the constants from the physical description of the spindle error motion, and from the measured data are similar. Results for each of the three tests in Figure 1 show theory and experiment in agreement. R_c was calculated from measured data of tests #1 & #2 to be 19.9 mm and 20.2 mm respectively vs. 21.2 mm as calculated from the physical construction.

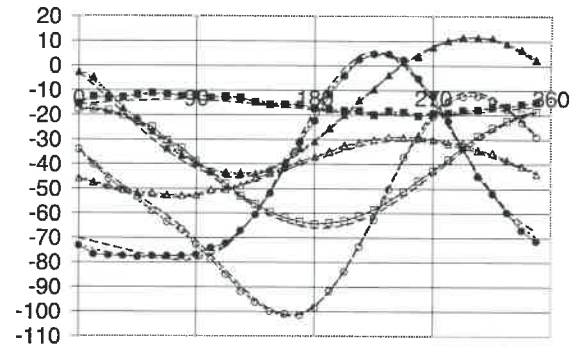


FIGURE 1. Orthogonal positions of artifact [mm] vs. angular orientation [degrees] for tests: #1(squares), #2(triangles), & #3(circles). Comparison of measured results (markers), calculated from knowledge of physical system (long dashes), and calculated from analysis of measured data (short dashes). X & X' markers are outlines, Y & Y' markers are solid.

CONCLUSIONS

Equations were presented that define synchronous spindle motion and measurements for both fixed and rotating sensors. A method of compensation for fundamental radial error motion, and insight into artifact centering were provided. An alternative definition for residual motion was proposed. For the three tests conducted, measured results agree with theory.

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DIAMOND TOOL WEAR MEASUREMENT BY EBID

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INTRODUCTION

Accurate measurement of the wear of diamond tools is of great interest for precision machining. This is often done indirectly using surface finish measurements, combined in some studies with tool force measurements [1]. However, a proper investigation of the tool wear, suitable for the development of wear models, requires the quantitative characterization of the diamond tool edge radius and the flank wear length as a function of the cutting distance for given machining conditions and materials. It is difficult to obtain the sectional curve of the diamond tool edge because the edge radius is on the order of a few tens to hundreds of nanometers and the diamond itself is nonconductive. Up to now, only a few results have been reported for measurement of the edge radius, and these varied due to limitations of the methods. Atomic force microscope (AFM) and AFM combined with nanoindentation were used to map the tool edge topography [2, 3]. The radius of commercially available AFM probe tips is often on the same order of the diamond tool edge radius. Errors arise from the convolution of the probe tip and tool edge profile and the elastic spring back of the indented material.

With the capability of high resolution imaging, the scanning electron microscope (SEM) is desirable for direct imaging of diamond tool edge sharpness. An SEM, which was specially configured with two secondary electron detectors, was used to obtain the profile curve of the diamond tool [4]. The cutting edge profile was determined by comparing the signal differences in two SEM images, which are affected by the convex and the concave shape of the sample surface. Measurement errors come from the reliability of the empirical formula to determine the inclination angle of the diamond tool. Gold thin film coated on the diamond tool

surface to eliminate charging will also affect the measurement accuracy. To improve image contrast and obtain quantitative topographic information in direct SEM observation of the diamond tool edge, Drescher developed a simple technique applying the contamination line in a conventional SEM [5]. In this paper, the electron-beam-induced deposition (EBID) method for tool-wear measurements has been further developed and improved using a state-of-the-art SEM instrument. EBID is a phenomenon that is present during SEM observations. The electron beam will crack the hydrocarbon contaminants present in the vacuum column to form a cross-linked polymer [6]. Under proper operating conditions, an EBID line can be deposited across the rake and flank face of an uncoated diamond tool, which allows for direct observation of the tool edge and wear flank geometry.

EXPERIMENT SETUP

A 45° sample holder was used in the experiment to set the diamond tool edge exactly perpendicular to the electron beam. Figure 1a is the side view of the diamond tool on a 45° sample holder. After the SEM electron gun and apertures were aligned and the astigmatism was removed, scan rotation was selected to bring the cutting edge vertical on the SEM CRT screen. The center area of tool edge was located and the scanning mode was switched to line scan. After several seconds, a hydrocarbon EBID line was found to be deposited on the surface (Figure 1b). To get the profile of the EBID line on the edge, the diamond tool edge was tilted 45° towards the detector (Figure 1c). SEM images of the tool edge with the EBID line on it were collected at different magnifications. Figure 1d is a SEM image collected after the 45° tilt operation.

All measurements of the diamond tool edge by EBID method were done using a JEOL JSM-6400F field emission SEM. After careful analysis of the factors involved with the fabrication of EBID in a controlled manner on a non-conductive surface, a small range of SEM conditions were determined to produce satisfactory EBID lines on the diamond tool. A series of experiments were run to determine the optimum conditions from the range considered. The best operating conditions were obtained with a beam energy of 2 keV, a condenser lens setting of 6, and a 30 μm objective aperture. The highest magnification of diamond tool edge obtained using EBID method was 100,000x. During the measurement process, the sample was tilted only once (45° to view the edge profile). Compared to the earlier work at the PEC [5], mechanical rotation of the sample stage was replaced by changing the direction of the electron beam scanning. Since the mechanical change of the sample stage may bring about some error, tilting the diamond tool only once will result in more precise measurement.

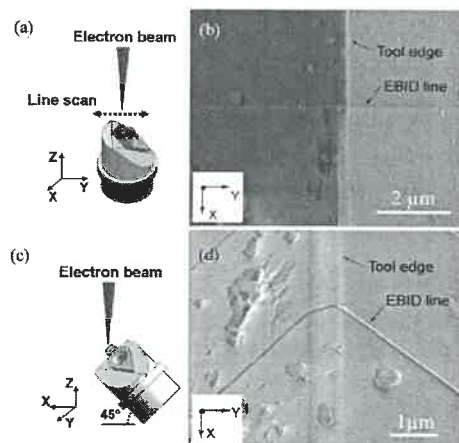


FIGURE 1. Illustration of diamond tool position in SEM chamber and measurement steps. (a) Side view of the diamond tool fixed on a 45° sample holder; (b) EBID line deposited across the center of the tool edge; (c) Diamond tool edge is tilted 45° towards the detector; (d) SEM image of diamond tool edge with EBID line collected after 45° tilt operation.

SEM IMAGES OF DIAMOND TOOL EDGE

Figure 2 is the SEM image collection of tool edge for both new and worn diamond tools. Figure 2a, b are the cutting edge images of a new diamond tool with an orthogonal cutting flat

nose. Figure 2c, d and Figure 2e, f are the worn tool edge images after cutting 7.5 km of 6061 Al and 15 m of 1215 steel, respectively. EBID lines separated several micrometers away from each other at the tool edge center have exactly the same profile. In the low magnification images, the length of the EBID line is about $7\ \mu\text{m}$ on each side, which would capture the whole profile of interest. In the higher magnification images, the edge sharpness can be observed in better detail. By comparing worn and unworn tool edge images, it is seen that the unworn tool edge profile is much sharper than that of the worn tool. For the worn tool edge images, worn patterns of the tool edge for cutting the two materials are quite different.

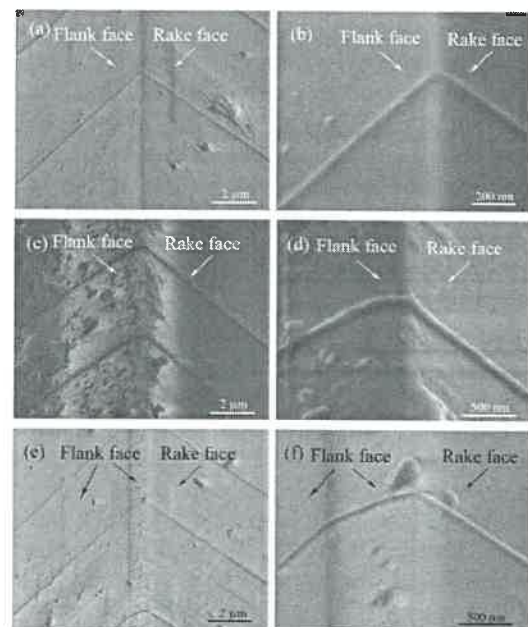


FIGURE 2. SEM images of diamond cutting tool edge with flat nose. (a), (b) Cutting edge of new diamond tool; (c), (d) Cutting edge of worn diamond tool after cutting 7.5 km of 6061 Al; (e), (f) Cutting edge of worn diamond tool after cutting 15m of 1215 steel. Magnification: 10,000x for (a), (c), (e); 50,000x for (d), (f); 100,000x for (b).

SEM IMAGES ANALYSIS

During the EBID measurement process in the SEM, the diamond tool edge has been tilted 45° , so the tool edge in the SEM images in Figure 2 has the same tilt angle. To get the real edge geometry, the image used for analysis should be stretched 141% ($1/\cos 45^\circ$) in the vertical direction to compensate for the other 45° tilt.

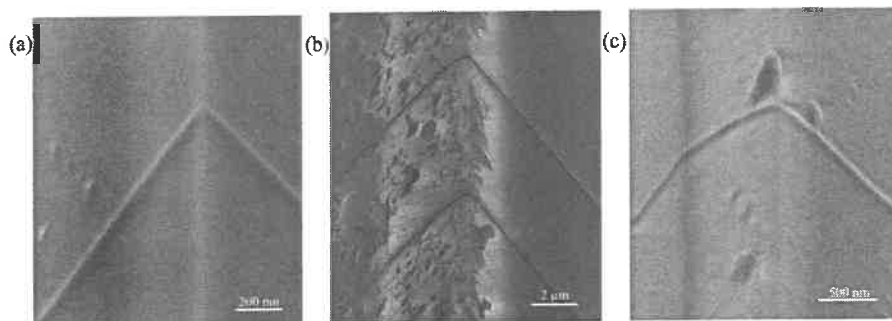


FIGURE 3. SEM images stretched 141% vertically from Figure 2b, c and f, respectively.

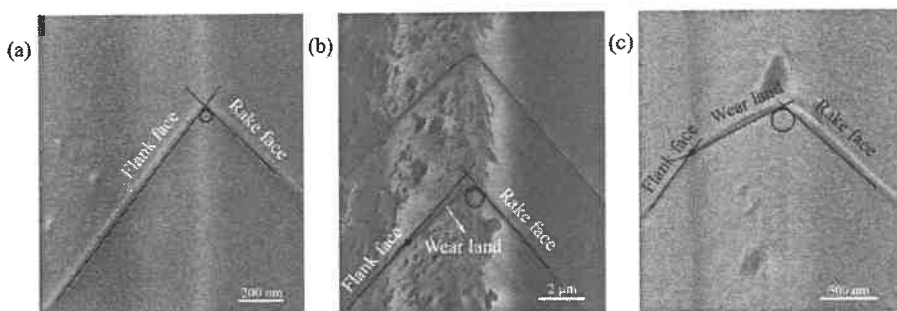


FIGURE 4. Matlab program fitting process for (a) new sharpened diamond tool, (b) worn tool after cutting 7.5 km of 6061 Al, and (c) worn tool after cutting 15 m of 1215 steel.

Figure 3 show the SEM images stretched using Figure 2b, c and f. Measured from the stretched image of new tool edge, the angle between the EBID line on rake face and flank face is about 84° , which is in agreement with the 6° clearance angle specified by the tool vendor. It can be concluded that the tilt and stretch operation is reliable for obtaining the real diamond tool edge geometry.

To quantitatively analyze the edge radius, a Matlab program was developed to process the digital images. After drawing a set of points along the tip area of the edge, a circle based on a least square method can be fit to the points to describe the tool edge radius. The program will count the pixel value of the circle radius. By comparing the pixel value with that of the scale of the images, the actual value of the edge radius can be obtained. While the circle fitting process is partly subjective, gradually drawing points very close to the tip area of the cutting edge will allow a best-fit circle to be found. To get the edge radius of the new diamond tool, the highest magnification (100,000x) image was used and stretched. Two lines were drawn close to the EBID line to delineate the tool edge. The image was then expanded to better view the

tool tip, and points were selected that correspond to the tool edge radius. Note that the circle describing the edge radius needs to be tangent to the rake face and flank face trace lines. Figure 4a shows the best fit circle to describe the edge radius of a new diamond tool before cutting.

The wear pattern for a worn tool is quite different for the two workpiece materials. For the diamond tool cutting 1215 steel, the wear land can be seen clearly and measured directly. Figure 4c shows the fitting result for a worn tool after cutting 1215 steel. For diamond tool cutting 6061 Al, the wear land is not as obvious, and the initial line on the new tool image is needed to make a comparison. First, the initial edge geometry formed by two straight lines on the rake and flank face in the unworn tool image was obtained as the base line. Then, the base line is transferred to the worn tool image. The deviation from the base line is used to describe the flank wear land and to fit a circle to evaluate the edge radius. Figure 4b is the fitting result for a worn tool after cutting 7.5 km of 6061 Al.

WEAR RESULTS FOR 6061 ALUMINUM AND 1215 STEEL

To demonstrate the EBID measurement method for diamond tool wear, two cutting experiments are discussed. Two workpiece materials were used. The diamond cutting tools are flat-nose with 6° clearance angle, which are supplied by Chardon Tool and Supply Company. The orthogonal cutting was performed using the ASG2500 Diamond Turning Machine. Four 6061 Al disks with the total 10 km cutting distance were prepared to study the gradual edge retraction of the diamond tool. The diamond tool was measured using the EBID method after each test. In the case of 1215 steel, a total cutting distance of 20 m was used. Cutting parameters and diamond tool conditions in the two experiments are shown in Table 1.

Table 2 gives the tool wear condition corresponding to Table 1. The edge radius and wear land length increased with the cutting distance. The edge radius of the new diamond tool is about 20 nm, which is the same as the value reported by Asai [4] and much smaller than the value reported by Lucca [2]. Note that the wear rate for 1215 steel is at least two orders of magnitude larger than that for 6061 Al. This is consisted with the chemical wear effect proposed for ferrous metals [7].

TABLE 1. Orthogonal cutting experiment parameters and tool conditions.

	Experiment 1	Experiment 2
Workpiece	6061 Al disk Thickness: 0.81 mm	1215 steel disk Thickness: 1.20 mm
Cutting speed	3 m/s	2.13 m/s
Depth of cut	2 μ m	1 μ m
Cutting distance	2.5, 5, 7.5, 10 km	10, 15, 20 m
Oil and gas	Yes	Yes
Diamond tool	Flat nose Top rake: 0° Clearance angle: 6° Cutting edge length: 2 mm	

TABLE 2. Wear conditions of diamond tool for cutting 6061 Al and 1215 steel.

		Edge radius (nm)	Wear land length (μ m)
Experiment 1	New tool	25	0
	Al_2.5 km	142	1.20
	Al_5 km	168	1.96
	Al_7.5 km	297	2.99
	Al_10 km	299	4.45
Experiment 2	New tool	23	0
	Steel_10 m	75	0.53
	Steel_15 m	110	0.87
	Steel_20 m	132	0.88

CONCLUSION

The EBID method was used to measure the diamond tool edge sharpness in a conventional SEM instrument. It is a direct and effective way to obtain tool edge radius and wear land without coating on diamond surface. The edge radius for new commercial diamond tool is around 20 nm, somewhat smaller than values previously reported. Because of the higher magnification of the diamond tool edge images, this method is considered to be the most accurate measurement to date. Based on the measurement results of the edge radius and wear land, the wear volume can be estimated and wear models can be developed. In future work, the EBID method can be further refined to get higher magnification images for improved data fitting.

ACKNOWLEDGEMENTS

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NANO IMPRINTING SYSTEM OPERATED BY MULTIPLE MICRO ROBOTS

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1. INTRODUCTION

For these years, the new nano and micro imprinting technology has been more important to provide the nanoscale accurate 3D patterns onto such polymeric samples with low cost. In the conventional process, the ultra precise pressing machine that is equipped with many fine positioning facilities can be employed to push the nano or micro scale 3D molds to the target. However it's still so expensive to maintain such ultra precise machinery. In our group, the unique micro robot that consists of the piezo elements and the electro magnets have been developed and it succeeded in providing the microscopic indentations with accurate space of a few micrometer. Here the diamond tool of 0.5 micron radius can be indented on to the surface with the mechanical resonated cantilever on the micro pizo driven robot. In the primary experiments, it is demonstrated that the accurate array of many micro indentation can be given into the specified location.

In this report, it is described that this small piezo driven robot with such the micro indenting tool has been improved to provide such micro and nano imprinting application, there another metrology system and fine manipulator also have been developed to coordinate its position precisely and to press such micro mold onto the target surface. Here the small robot location can be monitored by the micro optical linear scales to make the precise positioning with the resolution of 50nm and the vertical manipulator that is driven by the simple voice coil motor is equipped for imprinting the nano and micro molds. Also the micro optical fiber based sensor is attached to detect the tool displacement to check the depth of tool. This simple micro robot technology can allow to provide such nano-imprinting system in the SEM chamber with the real time observation performance.

2. EXPERIMENTAL SETUP

2.1 Nano indenter on piezo driven micro robot

As shown in Fig.1 and Fig.2, in our previous reports, the unique micro robot that consists of the piezo elements and the electro magnets have been developed and it succeeded in providing the microscopic indentations with accurate space of a few micrometer as shown in Fig.1 and Fig.2.

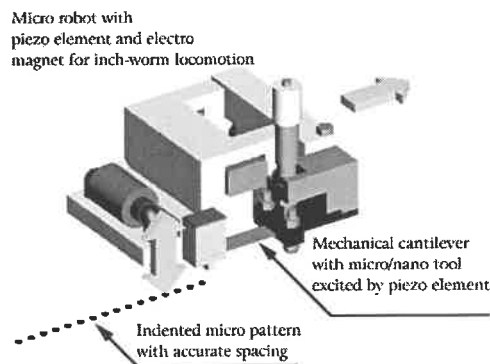


FIGURE.1 Piezo driven micro robot with micro-indenter on resonating cantilever

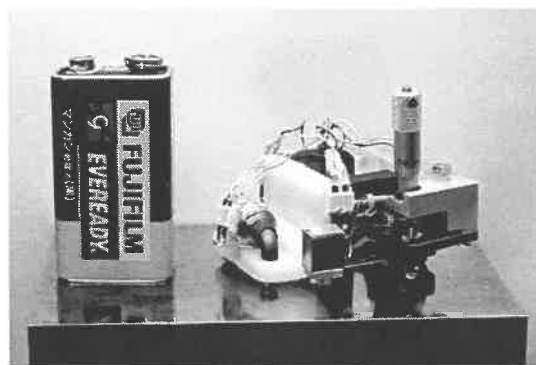


FIGURE.2 Developed micro robot with micro-indenter on resonating cantilever to provide accurate micro indentation

This structure allows to move precusely on a plane with the manner of inchworm[3]. The minimum step width of this inchworm is ranging from 10 nm to several um and the driving speed above several mm per second can be performed. The overall system as illustrated in Fig.3, can control these small robots under the microscope vision based navigation property.

2.2 Mechanical cantilever for micro indentor excited by piezo element

In order for micro robots to carry out micro patterning process, it requires to make a tiny device that has simple mechanism. So, we focused on using the cantilever resonance for main principle. Fig.4 shows the structure of micro processing mechanism using cantilever resonance. This mechanism consists of Height adjuster parts, piezo element which generates a vibration, cantilever, tool head, and micro tool. When piezo element is put an alternative current, it can start to generate the vibration of tool head by transmitting its small oscillation to cantilever. If this frequency is same as the resonance frequency depending on the cantilever system, the tool head will respond with high amplitude of vibration. We can estimate the resonance frequency by the simple kinematics as well known Rayleigh method. Typically parameters to estimate that value are shown in Fig.5. Then we can get the mechanical resonance frequency of this mechanism as 33.6Hz. Fig.6 shows such actual vibration characteristic that indicates the resonance frequency of 32.1Hz.

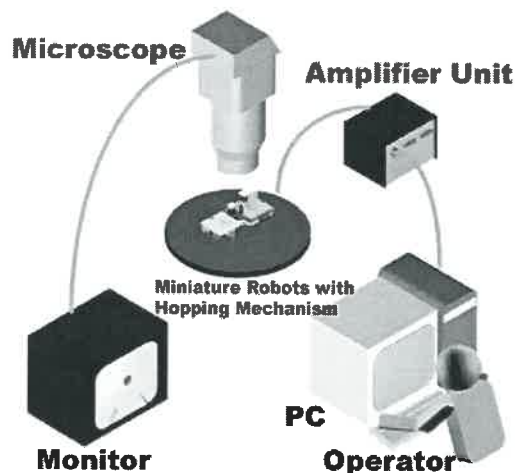


FIGURE.3 Overall system that can control the small robots with micro tool under the microscope

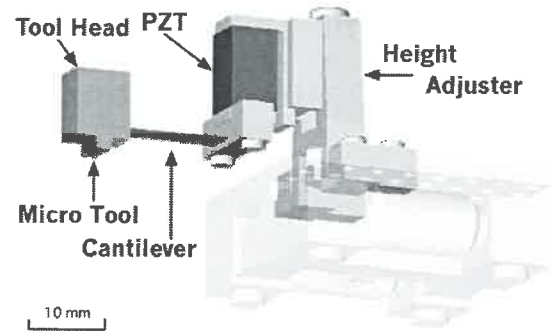
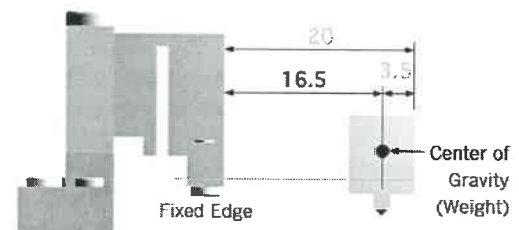


FIGURE.4 Micro indenter on cantilever resonated by piezo element



Length of beam	$l = 16.5 \text{ mm}$
Width of beam	$b = 5 \text{ mm}$
Thickness of beam	$t = 0.1 \text{ mm}$
Young's Modulus	$E = 200 \text{ GPa (Stainless Steel)}$
Density of beam	$\rho = 7860 \text{ kg/m}^3$
Mass of weight	$m = 1.0 \text{ g}$

FIGURE.5 Dimention of piezo excited cantilever and micro indenter

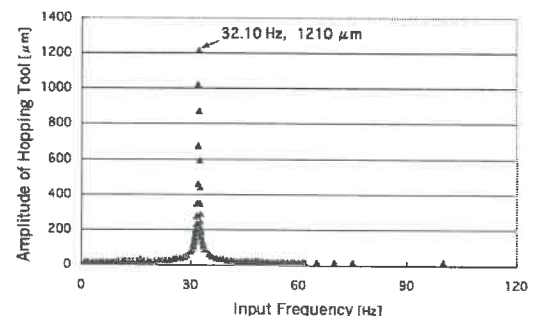


FIGURE.6 Dynamic performance of cantilever and maximum amplitude of vibration

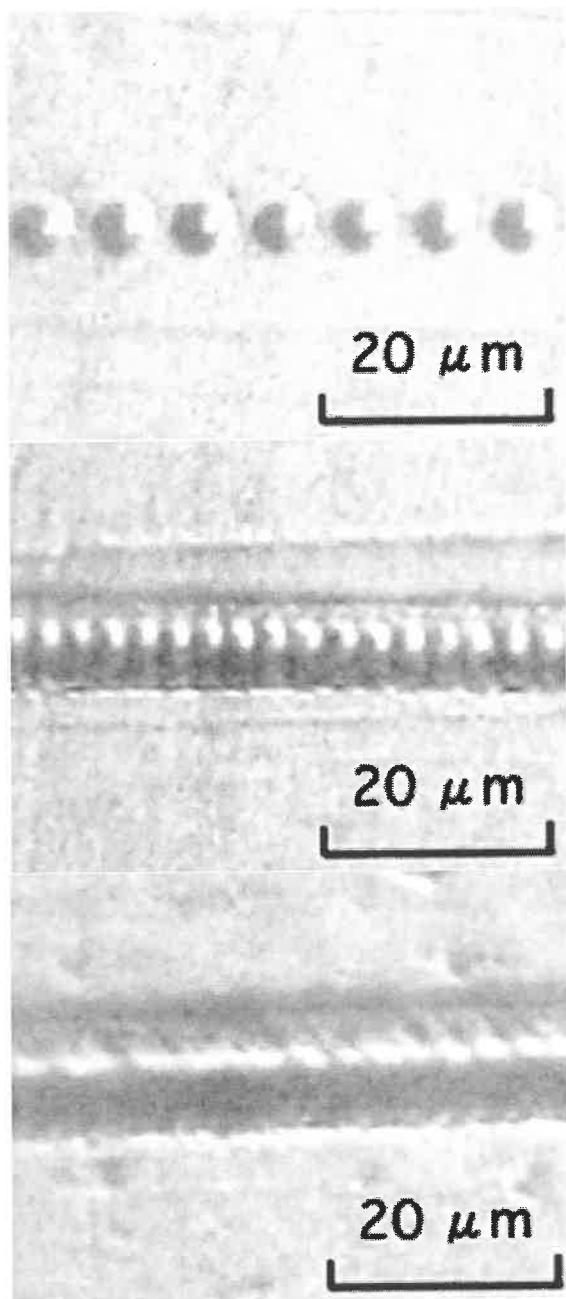


FIGURE.7 Several linear micro indentation pattern by the small robot with the hopping indenter.

2.3 Experimental results of micro surface machining

As the primary experiment to check the basic performance, the micro surface machining process with hopping micro indent was carried out. The diamond tip of 0.5 μm was attached at the end of indenter and the sequential indenting with the accurate space can be given onto the target workpiece of aluminum material by synchronizing the tool indentation and small robot locomotion. The typical results are shown in Fig.7, here several different step width of the inchworm locomotion by the single small robot are given so that it can provide the unique linear patterns. The accurate indentation pattern can be machined on the target although it is maneuvered by the open loop control.

3D topography measured by the laser scanning microscope is shown in Fig.8, when the micro groove is machined onto the surface with the smaller stepping locomotion. Then such micro indentations can be overlapped to get unique micro patterns.

In order to expand the performance as to provide 2 dimensional micro surface patterning, two micro robots, one of which has the hopping indenter and the other has the sample holder on its top also are developed as shown in Fig.9. Two small robot can be controlled to move X and Y directions relatively. Typical result is shown in Fig.10, here the array of micro dimples with the diameter of 3 μm are achieved precisely although these small robots are controlled sequentially by the open loop property.

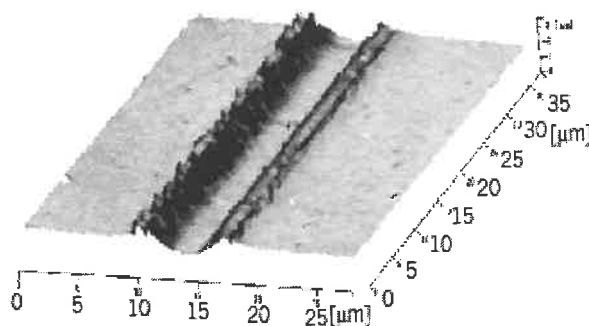


FIGURE.8 Micro groove machined on the surface with the overlapped micro indentation by smaller stepping locomotion

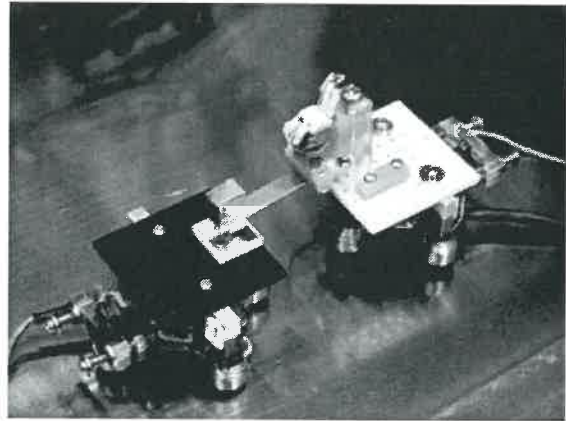
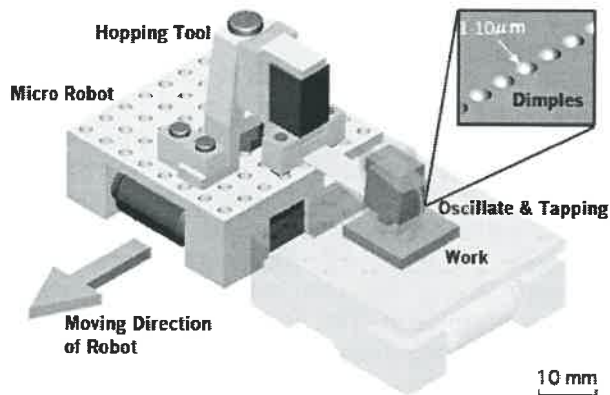


FIGURE.9 Combination of micro robots with hopping indenter and sample plate holder can provide unique micro surface patterns

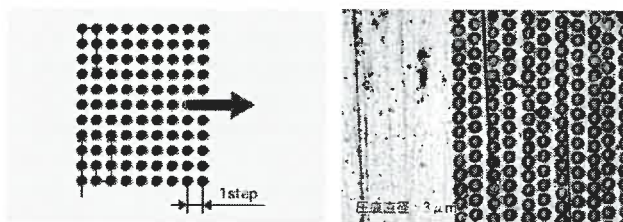


FIGURE.10 Accurate array of the micro indentation when two small robots are controlled to move to go back and forth

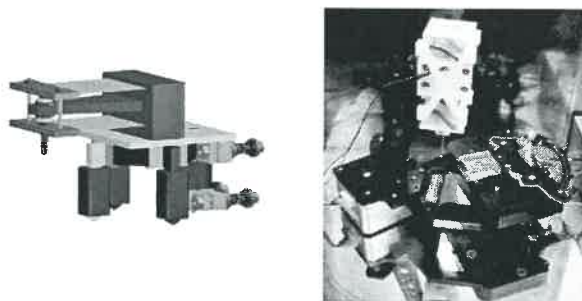


FIGURE.11 Nano mold attached to the voice coil motor driven leaf spring head on micro robot

3. NANO IMPRINT EXPERIMENT

In the previous section, micro single indentation patterning system organized by micro robots and resonating tool was proposed. Here the imprinting system are developed to achieve much more smaller pattern. As shown Fig.11, the imprint head is supported by parallel spring, and this parallel mechanism can be controlled by simple voice coil motor, which is implemented on the micro robot. In the experiment, the nano pattern mold is attached

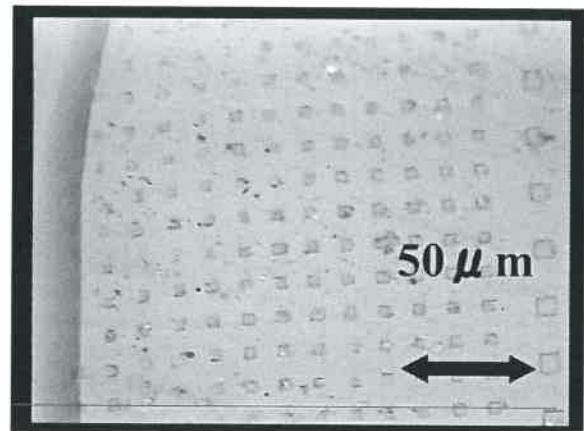


FIGURE.12 Imprinted nano pattern of UV cured resin

onto the tip end of the imprint head and it can be pushed on UV cured resin. After UV light is illuminated over the mold, then nano pattern can be printed precisely.

4. CONCLUSION

In this report, unique nano imprinting system organized by micro robots was discussed. At first, the single point indenter was driven by the piezo excited cantilever on the micro robot so that it can provide series of micro indentation pattern. As the second part, the nano mold was attached at the tip end of the voice coil motor driven parallel spring on the micro robot. Then UV cured resin can be molded after UV light is given over the target. We are on-going development with the real time process control and observation system in SEM chamber.

NEW MECHANICAL PRETREATMENT PROCESS FOR THERMALLY SPRAYED ALUMINUM CRANKCASES

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INTRODUCTION

The increasing demand for vehicles with reduced fuel consumption and emission is the reason why the material aluminum is frequently used for crankcases in the automotive industry. A method to increase the wear resistance of the cylinder bore surface is to use hyper-eutectic cylinder liners, which are cast-in or are assembled by force fitting. Another possibility to improve the tribological behaviour is to coat the surface with a wear-resistant coating by thermal spraying. To guarantee functional and heavy duty coatings, it is necessary that there is a strong bond between the coating and the aluminum body. To achieve this aim, the surface of the cylinder bore has to be pretreated. State-of-the-art are blast processes as well as employing chemical bond coats. These processes have some disadvantages like high investment and process costs, environmental pollution with chemical remainder or expensive grit reconditioning. Therefore, a new pretreatment process has been developed.

This new machining process is a kind of modified fine boring process. During machining a defined profile is cut into the surface by using a PCD-tool with a special geometry. The tool rotates and moves axially with a defined feed rate through the cylinder bore. The resulting profile is a reproducible and defined structure which — regarded in a cross-section — reminds of a dovetail. The undercuts in the cylinder surface secure a strong bonding between the aluminum substrate and the coating. Adhesion tensile strengths up to 60 MPa were measured, which clearly exceed the values which can be achieved by high-pressure water-jet roughing. Another advantage of the mechanical pretreatment method lies in the fact that the machining can be accomplished on the fine boring machine. The fine boring process is a step in the process chain which cannot be substituted up to now. Therefore, there is no necessity for an extra pretreatment machine. This also means that additional time and therefore costs for the transport and a readjustment in the fixture of the crankcase can

be avoided. Another advantage of this new machining process is the short cutting time with about 10s per cylinder bore, depending on material and cylinder length. Therefore, the process can be excellently integrated into the process chain.

The significant advantages of this new pretreatment process seem to offer an interesting alternative to the state-of-the-art pretreatment processes for thermally sprayed light-metal crankcases.

STATE-OF-THE-ART

Thermal spraying is an excellent method to provide cylinder bores of aluminum crankcases with a wear resistant coating. This coating has a thickness of one or two hundred microns after the concluding honing process. To assure the functionality of the coating on a real engine it is necessary that there is a strong bond between the aluminum alloy and the spraying material. The main reason why the coating adheres to the substrate is mechanical interlocking. Besides, the adhesive tensile strength is influenced by adhesive forces and diffusion processes between the different material components [1]. State-of-the-art are blast processes like high-pressure water-jet roughing or grit blasting, as well as employing chemical bond coats like the NiAl/Flux procedure. These processes show profound disadvantages like high investment- and process costs or the laborious disposal of the chemical remainders.

These disadvantages were the reason to look for an alternative pretreatment method. This process should be a cutting process with a defined cutting edge. Furthermore, it has to meet the following demands:

- High adhesive tensile strength between aluminum and coat (> 30 MPa)
- Machining time < 30 s / cylinder bore
- Defined and repeatable surface topography

- Applicable to the fine-boring process of the cylinder bore

The mentioned machining time and the adhesion tensile strength of more than 30 MPa are values which are known for the high-pressure water-jet roughing.

MECHANICAL ROUGHING OF CYLINDER BORES

After casting the crankcases, a lot of machining operations (drilling, milling) have to be carried out before the spraying of the cylinder bore is possible. One of these processes is the fine boring process which is normally accomplished on a machining centre. Therefore, the mechanical roughing process should be applicable to the fine-boring process. This implies that there is no need to change the clamping device and no blasting plant is needed so that investment and prime costs can be reduced.

The used tool body is a fine-boring tool. Only the cutting insert is replaced by a special insert for mechanical roughing.

During the cutting process the tool rotates and moves axially with a defined feed rate through the cylinder bore. The result is a helix with the cutting profile. During preliminary inspections it was shown that a mechanically roughed surface has to feature numerous undercuts, so that the coating adheres to the aluminum [2]. A lot of surface profiles have been tested and best results were gained with a profile which reminds of a dove tail (fig. 1). The two undercuts which are arranged symmetrically inhibit the removal of the coating by occurring shear forces.

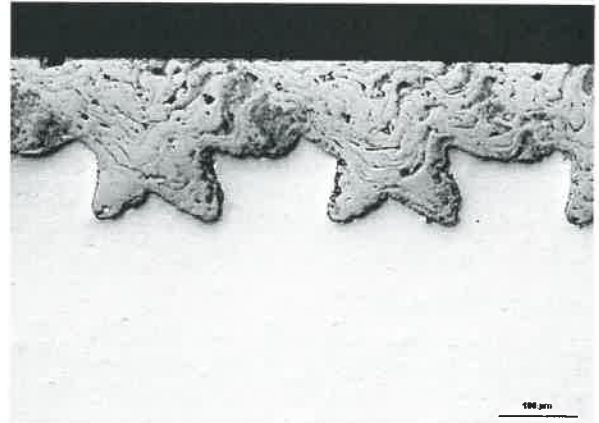


FIGURE 1. Dove tail profile

Figure 2 shows a schematic diagram of a roughed cylinder bore. Different cutting materials (carbide tools, coated carbide tools, CBN) were examined, at which polycrystalline diamond (PCD) proved to be the best choice for roughing aluminum-alloy crankcases.

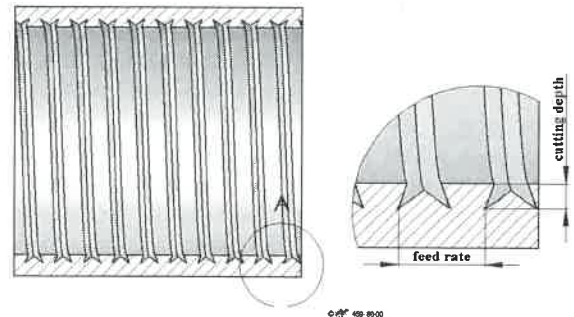


FIGURE 2. Mechanically roughed surface

As a result of the adhesive tensile strength test, values up to 57 MPa were measured. Best results were reached when the cutting depth amounted from 100 to 120 μm . When the cutting depth is too low ($<75 \mu\text{m}$), the undercut area is not sufficient and the bond between the coat and cylinder bore is weak (fig. 3). A further increasing of the cutting depth is not advisable because the strain to the tool increases. Furthermore, more spraying material is needed to pour in the undercuts. This causes higher spraying times and increasing material costs.

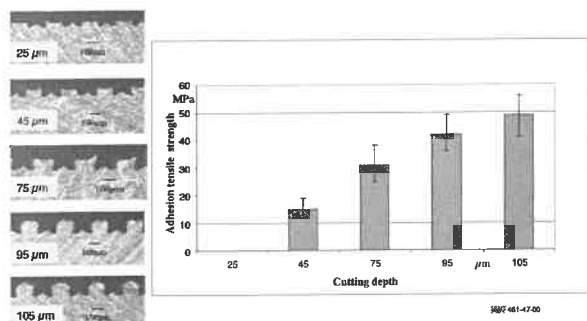


FIGURE 3. Correlation between cutting depth and adhesion tensile strength

The tool geometry was developed at the Institute of Machine Tools and Production Technology in Braunschweig, Germany. The tool consists of two parts: a carbide base body and a soldered PCD cutting insert. The insert shows two edges each with a corner radius amounting to 20 μm. The geometry is produced by wire electro discharge machining and the chosen edge and entering angle affect the shape of the undercuts.

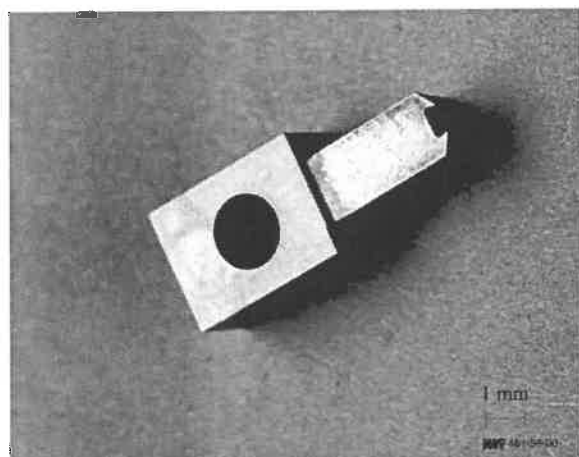


FIGURE 4. Double edged PCD tool

To understand the cutting process figure 5 shows a schematic diagram. During machining the first edge produces the first undercut which looks like a sawtooth. Then, the second cutting edge generates the opposite undercut so that the dovetail profile is completed (fig. 5).

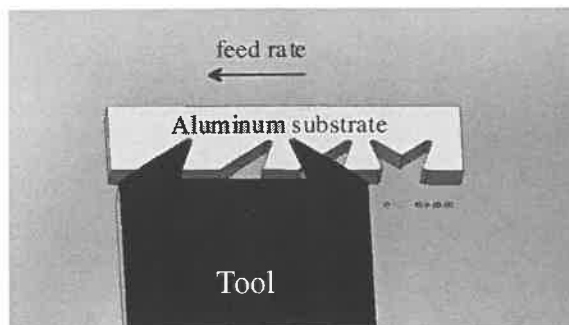


FIGURE 5. Schematic diagram of the cutting process

The first experiments with this tool were dry machining processes. In the process sometimes the chip space was loaded completely, especially in machining very ductile aluminum alloys with a low amount of silicon (0.5 %). In that way there was no possibility to produce the roughing profile. These adhesion problems can be completely avoided by using conventional coolant (emulsion).

The average cutting time is about 10 s per cylinder bore (dependent on bore length and material).

Summary

This new mechanical pretreatment method uses a special PCD insert which is clamped to a fine-boring tool. This insert consists of two cutting edges which are arranged symmetrically. The cutting profile reminds of a dovetail. The roughing process shows some strong advantages in comparison to conventional pretreatment methods like high-pressure water-jet roughing or chemical bond coats. The adhesive tensile strength is high (up to 57 MPa) and the cutting time is short (~10s / cylinder bore). The biggest advantage is that the fine boring machine can be used so that there is no need for an additional pretreatment machine and for changing the clamping of the workpiece. This means that process- and investment costs can be reduced.

Therefore, the mechanical pretreatment method seems to be an interesting alternative to conventional pretreatment methods.

Currently, different modifications of the cutting tool and the influence of coolant are tested to improve tool lifetime.

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GRINDING HARD AND BRITTLE MATERIALS WITH CVD-DIAMOND MICROGRINDING WHEELS

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INTRODUCTION

Microcutting operations are distinguished from other processes like lithography by their high flexibility, short manufacturing time as well as the machinability of a broad spectrum of materials. Thus, they are very appropriate for the manufacturing of high-precision structures and components of microsystems technology [1]. This is shown by the example of coated single layer CVD-diamond grinding wheels.

This paper provides the results of novel CVD-diamond microgrinding wheels. The microgrinding process with diamond microgrinding wheels enables a fast and high-quality fabrication of microstructures in various hard and brittle materials [1, 2, 3].

CVD-coated tools for microgrinding operations were formerly only used for shaft tools, i.e. for peripheral grinding [4]. The advantages of CVD-coatings, which have been known for many years, had to be transferred to miniaturised grinding wheels [5, 6]. They have the same advantages as CVD-mounted points, but they can even be used with higher cutting velocities and higher material removal rates. Furthermore, according to their small grain size, hard and brittle materials can be machined with low edge breakouts when optimized machine parameters are used. This means that micro cracks and chipping can be minimized and better surface qualities can be achieved.

This paper is divided into two parts. At first, the tool manufacturing consisting of machining the cemented carbide tool body and the characteristics of the CVD-layer are described, and finally, the results of their application in machining hard and brittle materials are presented. The development of these tools offers new possibilities of microstructuring hard and brittle materials (fig. 1) [7, 8].

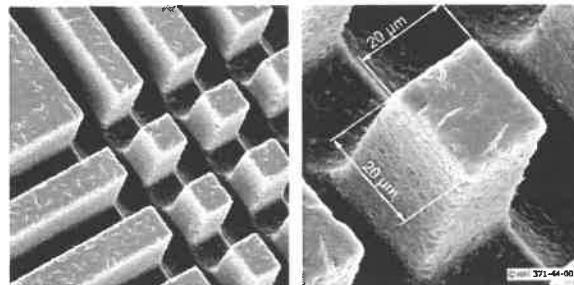


Figure 1: Examples of microstructured surfaces: Ground pillars in glass

TOOL MANUFACTURING AND SPECIFICATION

First of all, the configuration of the microgrinding wheels is described. They consist of a cemented carbide tool body and a rough CVD-diamond coating. After designing an appropriate geometry for these novel microgrinding tools (fig. 2), the inner diameter with fit size was ground. Afterwards, the tool body was fixed and oriented on a workholding device.

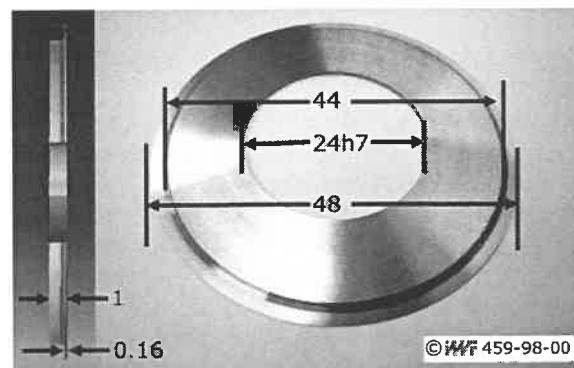


Figure 2: Geometry of the cemented carbide tool body

Subsequently, the cemented carbide tool bodies were produced with cup wheels, while the machine table was rotating. They have a diameter of 48 mm and a width of 0.16 mm. The design, which has already been tested, is a grinding wheel of the type "1A1". The tool bodies were machined at the Institute of Machine Tools and

Production Technology (IWF) and afterwards coated with a chemical vapour deposition (CVD) process at the Fraunhofer Institute for Surface Engineering and Thin Films. The CVD-diamond microgrinding wheels were analyzed with a scanning electron microscope (SEM). The diamond crystals have an average width of $5\ \mu\text{m}$ (fig. 3).

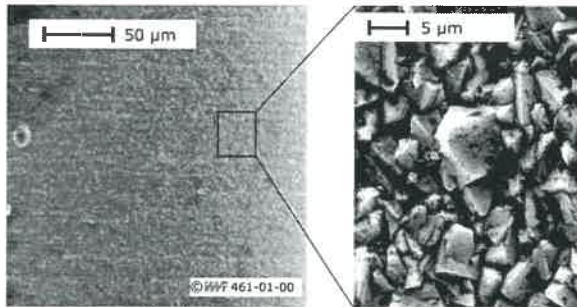


Figure 3: CVD-layer of the microgrinding wheel

Fig. 4 shows the cross section of a microgrinding wheel with its tool body and its coating. Also the thickness of the coating can be quantified as $10\ \mu\text{m}$.

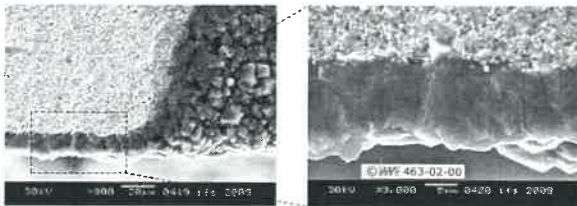


Figure 4: Cross section of the tool

EXPERIMENTAL RESULTS OF THE GRINDING TESTS

The materials aluminum nitride, aluminum oxide, glass, silicon and cemented carbide were ground with CVD-diamond grinding wheels, while the infeed as well as the cutting speed and the feed rate were varied. In addition to the in-process measurement of normal and tangential forces with a dynamometer during grinding, the edge breakout and the surface roughness of the ground of the groove were measured after the process with a confocal microscope. The wear of the microgrinding wheels was rated and documented by means of SEM-pictures of a new grinding wheel and after a defined material removal rate.

The analysis of the cutting forces showed that they were lower than 1 N for all investigated materials and process parameters. Figure 5 shows the development of the tangential and normal forces in dependency of the cutting forces by

grinding silicon. Both test series were run with the same specific material removal rate. As a result, it can be stated that higher feed rates lead to lower cutting forces.

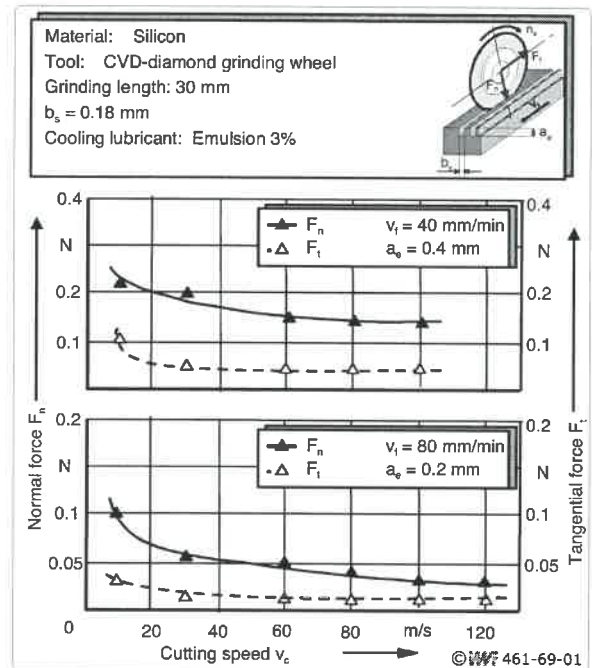


Figure 5: Cutting forces

The normal forces during the tests were approximately twice as high as the tangential forces. Moreover, it can be stated that the forces decrease when the cutting speed is increased. This is due to the fact that the equivalent chip thickness decreases for higher cutting speeds [3].

Figure 6 shows the development of the cutting forces in dependency of the related volume removed by microgrinding aluminum oxide. The normal and tangential force are relatively stable for the area of a specific material removal rate of $600\ \text{mm}^3/\text{mm}$ and afterwards there is a slight increase of the cutting force. These values can be used to rate the wear behavior of the microgrinding wheels. As there is only a small increase, the process and wear behavior of the CVD-diamond microgrinding wheel is satisfying. The process behavior of the tool is rated by SEM-pictures (fig. 7).

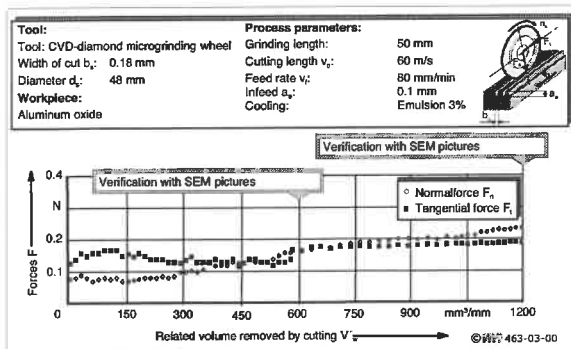


Figure 6: Cutting forces by microgrinding aluminum oxide

The material aluminum oxide was chosen for these tests, because it was the hardest of the investigated materials. For this reason, was expected that wear could occur. In the upper line, the topview of the peripheral surface of a new CVD-microgrinding wheel is shown for several magnifications. The crystalline, sharp-edged structure of the CVD-diamond grains is clearly visible. The pictures of the middle line show similar pictures after a specific material removal rate of 600 mm³/mm, the pictures in the lower line, the ones after a specific material removal rate of 1200 mm³/mm. The structure of the crystallites stays well preserved, and except for few adhesions of cooling lubricants and chips the configuration of the CVD-layer hardly changed. Only the small increase of the normal force for higher specific material removal rates by cutting suggests little wear.

Tool: CVD-diamond grinding wheel	Process parameters: Grinding length: 50 mm Cutting speed v_c : 60 m/s Feed rate v_f : 80 mm/min Infeed a_p : 0.1 mm Cooling lubricant: Emulsion 3%
Width of cut b_c : 0.18 mm Diameter d_c : 48 mm	
Workpiece: Aluminum oxide	

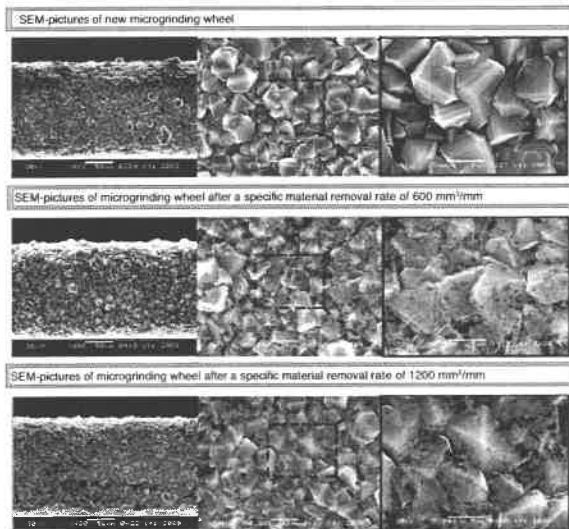


Figure 7: Wear behavior by microgrinding aluminum oxide

As further results of the grinding tests, it can be summarized that there was a maximum edge breakout of 5 μ m at the workpiece. They were 1 μ m for grinding cemented carbide and 5 μ m for grinding glass. The value of the edge breakouts is higher for up-grinding than for down-grinding and increases exponentially with higher cutting speeds [9].

CONCLUSIONS

Microgrinding with CVD-diamond grinding wheels is an appropriate method for microstructuring hard and brittle materials. In order to improve the machining quality when grinding these materials, new CVD-diamond microgrinding wheels were developed and the basic technological parameters were studied.

Their use by grinding silicon and aluminum nitride caused cutting forces lower than 1 N. Moreover, even at the hardest investigated material aluminum oxide, they did not cause visible wear, which means that they have a long tool life. Further advantages of these microgrinding wheels are low surface roughnesses at the flute base and marginal edge breakouts.

In addition to the mentioned materials, further hard and brittle materials can be used for microgrinding and -structuring operations. Based on the successful application of these tools, grinding wheels with convex and concave profiles are currently developed to broaden the spectrum of the machinable geometries.

Besides several different geometries of the tool body, the specifications of the coating will be varied. The focus will be on the ideal crystallite size for grinding operations. If they are too small, they do not offer enough chip space. On the other hand, if they are too large, they cause large edge breakouts at the workpiece and the surface roughness of the flute base will be too high.

The next aspect concerning these novel tools will be to regard the coatability of the tool bodies. The geometry of the tool body has to be chosen and machined in such a way that it can be coated reproducibly and that the coating does not strip.

ACKNOWLEDGEMENT

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STUDY OF TOOL WEAR AND CUTTING PROCESS OF ELLIPTICAL VIBRATION ASSISTED MACHINING

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INTRODUCTION

Studies have shown the benefits of applying micrometer scale vibration motion of a diamond tool during precision diamond turning (DT). Reported benefits include a decrease in machining forces and reduced wear of the diamond tool [1]. The reasons for decreased force and wear using vibration assisted machining (EVAM) are not well understood, nor is there a clear understanding of the tool wear process in normal DT operation for many materials.

Diamond wear is thought be due to abrasive or chemical wear or a combination of the two. Hard alloying inclusions, such as those in aluminum alloys, abrade the tool with increased cutting distance. Abrasive wear of diamond can be predicted by Archard's Law where the volumetric wear is proportional to sliding distance and the normal contact force. Mild steel has shown to produce high rates of chemical wear on diamond. Paul and Evans [2] predicted the chemical wear of diamond tools would be larger in workpiece materials that have unpaired d-shell electrons (pure iron has 4). They also noted that the resulting graphitization and/or diffusion of diamond into the workpiece would follow the Arrhenius equation, which depends on temperature.

The decrease in tool wear with EVAM has been widely noted, yet direct measurement is limited [1]. Cerniway [3] compared the wear patterns of standard DT and EVAM diamond tools in an SEM. He observed a beveled edge on the EVAM tool creating a large negative large rake angle but leaving a sharp edge that. This edge was thought to be the source of the excellent surface finish. Cerniway's experiments were conducted using W2 tool steel at 200 Hz operating frequency. Brinksmeier and Gläbe [4] measured wear after machining 1045 steel by standard DT and ultrasonic VAM. The worn tool edge was scanned using an atomic force microscope. A rounded edge was observed on the EVAM tool after machining 1045 steel at 40

kHz. The edge radius increased linearly with cutting distance. These two studies were conducted with different steel alloys, operating frequencies, and ellipse shapes. The following is a first report on a comprehensive study of diamond tool wear for conventional and EVAM machining.

MACHINING EXPERIMENTS

Aluminum 6061-T6 and 1215 steel were chosen for comparison because of their similar mechanical but dissimilar chemical properties. Al6061 contains hard alloying inclusions that contribute to abrasive wear, while the iron constituent of St1215 produces chemical wear.

TABLE 1. Properties of workpiece materials.

	Al6061	St1215
Density	2.7 g/cc	7.87 g/cc
Vickers Hardness	1185 MPa	1850 MPa
Young's Modulus	70 GPa	200 GPa

All cutting was restricted to orthogonal geometry with the workpiece narrower than the width of the tool (2 mm) as illustrated in Figure 1. All tools used were single-crystal, straight-edged diamonds with 6° clearance angle provided by Chardon Tools.

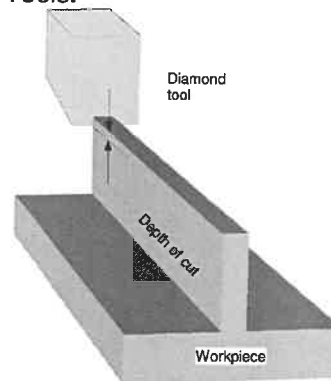


FIGURE 1. Geometry of cutting experiments.

Standard Diamond Turning

Standard cutting experiments were conducted on a two-axis diamond turning machine.

Replaceable Al6061 disks were held in a fixture mounted to the vacuum chuck while the circumference was removed by machining. St1215 was not available in sheets, so fins were machined into a St1215 cylinder. The cylinder was then turned in the same manner as the aluminum disks. TABLE 2 gives the cutting parameters for standard cutting.

TABLE 2. Standard DT cutting parameters.

	Al6061	St1215
Material Width	0.813mm	1.2mm
Depth of Cut	2 μ m	1 μ m
Cutting Velocity	3.4-2.66 m/s	2.13 m/s

Vibration Assisted Machining

EVAM machining was conducted using the piezoelectrically driven Ultramill mounted to a 3-axis Nanoform 300 diamond turning machine. This device enables elliptical tool vibrations with variable ellipse shapes and frequencies (1-4 kHz). Two piezo-electric stacks oscillate out of phase which drives the ceramic toolholder tip in an elliptical path. The shape of the motion is shown in Figure 2. The tool moves in an elliptical shape with period T while being fed across the workpiece surface with velocity, v . With EVAM, the tool is in contact with the workpiece for a longer distance than the upfeed during one cycle. Since wear depends on sliding distance, this factor needs to be calculated to compare EVAM with standard DT.

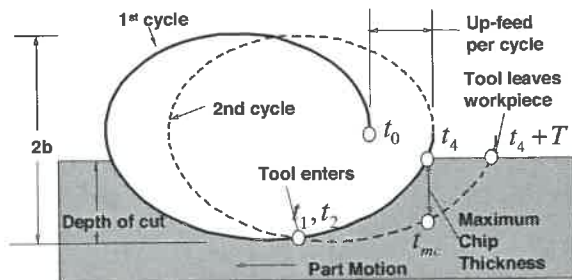


FIGURE 2. Cutout view of Ultramill device used for EVAM experiments.

The same depth of cut (2 μ m) was chosen for EVAM of Al6061 as with conventional machining with the same distance for upfeed. TABLE 3 gives the EVAM parameters. The minor axis and depth of cut are the same to observe if resultant chips are continuous or discrete. With the chosen depth of cut, workpiece velocity and ellipse minor axis, the maximum chip thickness is effectively half the depth of cut.

TABLE 3. Parameters for EVAM of Al6061.

Material Width	1.0 mm
Depth of Cut	2 μ m
Workpiece Velocity	2 mm/s
Operating Frequency	1 kHz
Ellipse Shape	a = 11 μ m b = 2 μ m
Upfeed	2 μ m
Sliding Distance / Upfeed	6.465 μ m/ μ m

EXPERIMENTAL RESULTS

After machining, diamond tool wear was measured in an SEM via electron beam induced deposition (EBID) [8]. A stripe of hydrocarbon contamination is formed along the cutting edge of a worn diamond tool by scanning a focused electron beam along the diamond surface. This stripe provides contrast to allow the cutting edge to be determined from the SEM image.

Tool Wear Observations

Figure 3 shows examples of EBID images obtained after standard cutting of Al6061 and St1215, and EVAM of Al6061. Tool cross sections directly traced from the EBID stripe show the general pattern of wear for the respective experiment. The standard cut Al6061 tool had an increasing edge radius and a wear land that was parallel to the uncut surface. The standard cut St1215 had an increasing wear land that formed at an angle with the uncut surface. The angle of this wear land remained the same with further machining. EVAM of Al6061 produced an increasing edge radius, but no wear land. Micro-chipping was also observed on the EVAM tool rake face. The micro-chipping became more severe, while the flank region of the tool remained smooth.

Volumetric wear from EVAM of Al6061 followed the same trend as standard cutting. Improvements in wear volume were not observed with EVAM. Figure 3 also shows progressive wear geometries measured from EBID images. The EVAM tool exhibited larger edge radii at less sliding distance than that for the standard cutting Al6061 tool. Wear geometries for the standard cut and EVAM Al6061 appeared to increase linearly. Wear geometries on the St1215 worn tool increased immediately after initial cutting, then increased at a more gradual rate.

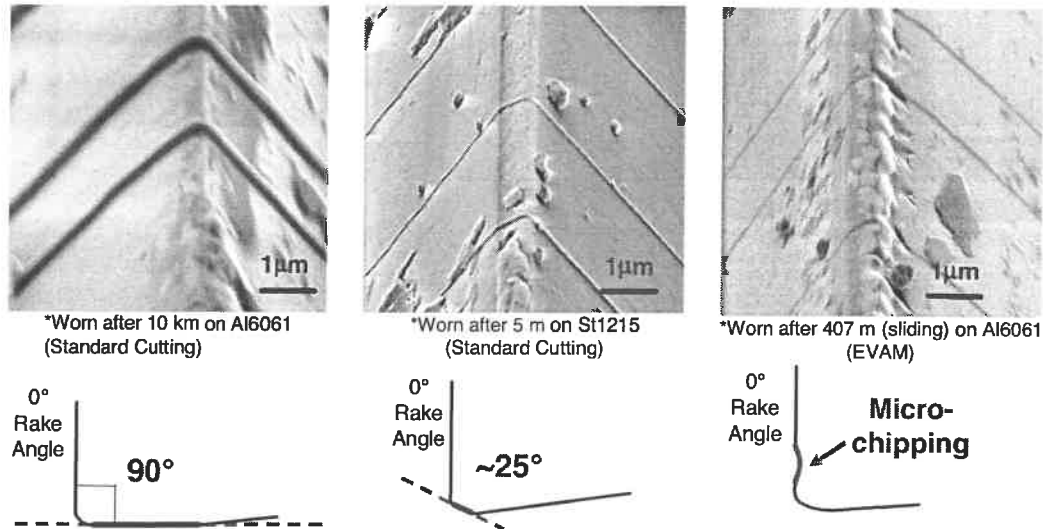


FIGURE 3. EBID images of the tool edge for conventional diamond turning of Aluminum 6061 (left), conventional machining of St 1215 (center) and EVAM of Aluminum 6061 (right). The bottom images are cross sections of the tool from the EBID images.

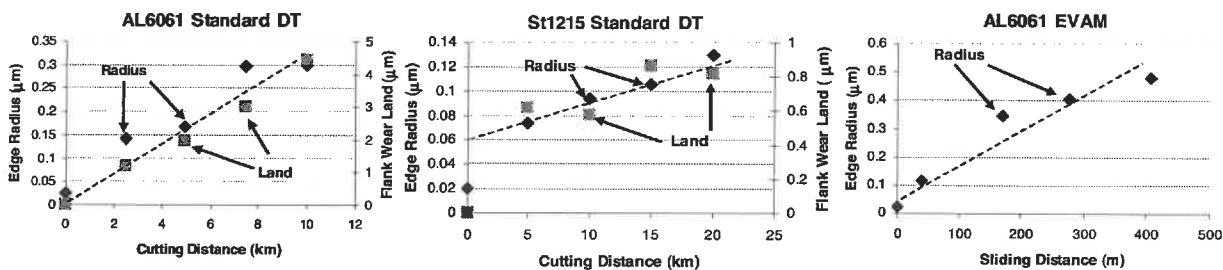


FIGURE 4. Measured tool wear geometry for the three experimental conditions.

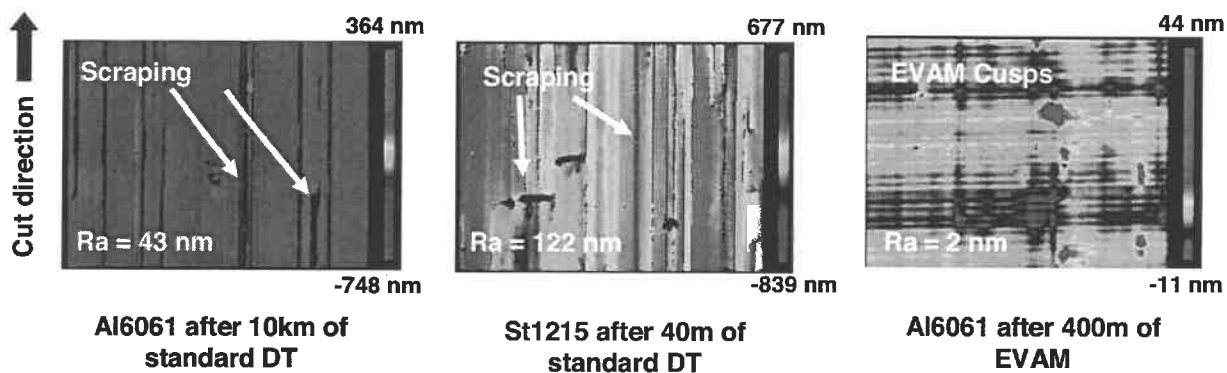


FIGURE 5. White light interferometer images of machined surfaces. The Ra roughness is shown in the lower left on each image. Scratching was reduced on the EVAM specimen.

Surface Finish

While the edge radius of the EVAM tool was greater than that for conventional machining, the surface finish was markedly better for aluminum workpieces. The surfaces generated by conventional machining showed scratches stemming from hard particles being dragged across the surface. This was not observed on

the EVAM surface. The St1215 surface exhibited similar scratching to the Al6061. Figure 6 shows the measured wear volume as a function of the sliding distance for two conventional and one EVAM experiments. The volume removed from the tool for conventional machining of steel was the largest. Similar wear for conventional machining of St1215 was

observed at 500x less cutting distance than that of Al6061. This higher wear rate is undoubtedly due to a chemical wear mechanism. A comparison of the conventional and EVAM machining of aluminum shows the same trend line fit to both sets of data. The sliding distance for EVAM is significantly larger than the cutting distance (650%) and is still much smaller than the cutting distance for the conventional machining experiments. Future experiments will provide data for EVAM machining on steel and other materials.

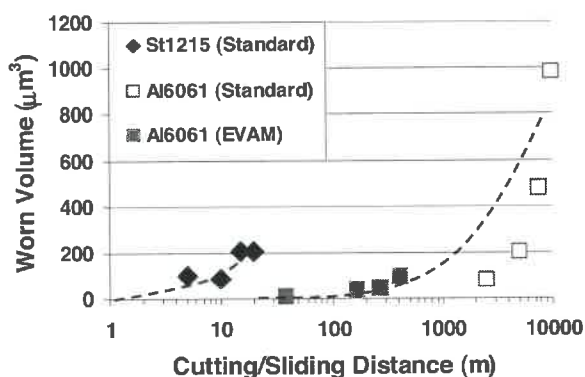


FIGURE 6. Worn volume of diamond tool as a function of cutting distance (standard cutting) or sliding distance (EVAM)

Cutting Forces

In addition to the wear studies described above, the cutting and thrust forces were measured during the experiments. The forces for conventional machining were larger for the steel than aluminum due presumably to the larger tool wear. The smallest was for the EVAM because the chip thickness is the smallest.

CONCLUSION

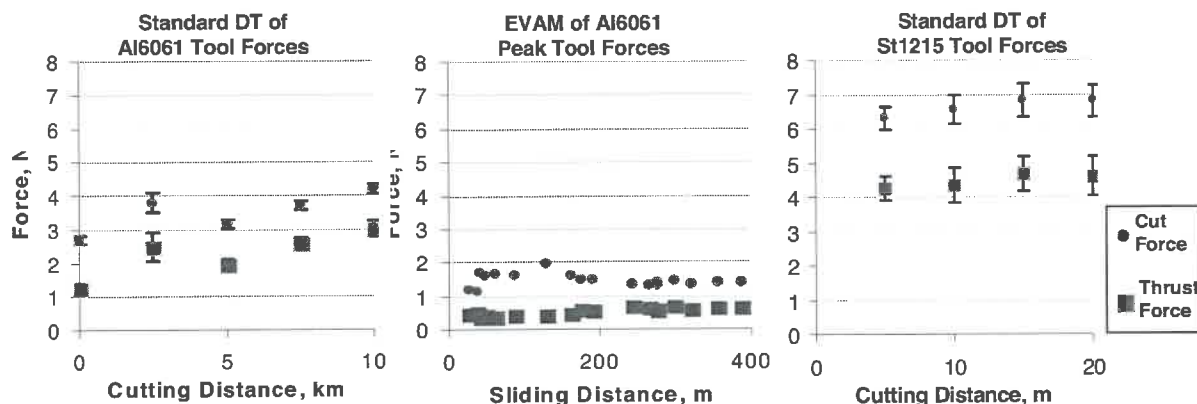


FIGURE 7. Cutting forces for conventional machining and EVAM for steel and aluminum workpieces.

Three cutting studies were conducted; standard DT of Al6061, EVAM of Al6061, and standard cutting of St1215. The EBID method was used to observe and measure successive diamond tool wear. The edge radius of the EVAM tool was the same or greater than the standard cutting tool at much less cutting/sliding distance on Al6061. Although benefits to tool wear weren't observed, a marked improvement in surface finish was observed due to the reduction in scraping. A baseline study on the DT of steel showed a much greater wear rate than that of aluminum. The worn tool shape was also different, with an upturned wear land. Surface finish of the steel surface was also exhibited scraping.

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PROPOSAL OF A SNAKE TYPE IN-PIPE MOBILE ROBOT USING PNEUMATIC ACTUATORS

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INTRODUCTION

We have many small diameter pipes that are cooling pipes for atomic power stations, boiler pipes, and gas or water pipe lines. They must be periodically inspected in order to protect the accident previously. Diameters and T-junctions of these pipes are different at the place where pipes change from the main to the branch. The inspection microrobot for these pipes must move different diameter and T-junction.

The in-pipe microrobot driven by wheels is very difficult to cross the step [1]. We have used cone-shape friction rings for the driving legs of the in-pipe microrobot. However, the microrobot driven by friction rings is also difficult to move in the pipe where diameter changes more than 3 mm [2].

Now, we propose a mobile microrobot that can surely move in T-junctions and a pipe whose diameter changes. The microrobot is constructed by the six rubber bellows as pneumatic actuator, six electromagnetic valves and six air feeding tubes. Outer diameter and natural length of the bellows are 16 mm and 108 mm, respectively. The bellows composes three somites by being arranged in two rows and three columns. The somite can make the arc like the bimetal by giving different pressure to each bellows. The arc somite can hold the pipe. The movement of the microrobot with

three somites is imitating as for the accordion movement that is the movement of the snake operation.

The fabricated mobile microrobot was confirmed to move in different diameter pipes and T-junction whose diameters are between 44 mm and 90 mm. Its speed was 25 mm/s.

STRUCTURE OF IN-PIPE MOBILE ROBOT

A structure of the fabricated robot is shown in Fig. 1 and photograph shown in Fig. 2.

The robot is constructed by the five rubber bellows as pneumatic actuator, five electromagnetic valves and five air feeding tubes. Outer diameter and natural length of the robot is 33 mm and 304 mm, respectively. The bellows composes three somite by being arranged in two rows and three columns. Outer diameter and natural length of the bellows are 16 mm and 108 mm, respectively. The somite is defined forward of the robot as first somite, second somite and third somite. Each somite can make the arc like the bimetal by giving different pressure to each bellows. The first somite and third somite are mechanisms that hold the pipe and the second somite is mechanisms that obtain displacement.

An experimental apparatus for measuring the characteristics of the robot is shown in Fig. 3. A

computer controls six electromagnetic valves through a valve controller. Three air feeding tubes are connected from the electromagnetic valves to each bellows of the robot. An air compressor is connected to the entrance ports of the electromagnetic valves and feeds pneumatic pressure to stretch the bellows. A vacuum pump is connected to the exit ports of the electromagnetic valves and feeds vacuum pressure to shrink the bellows.

MOVEMENT OF SNAKE IN A PIPE

We observed snake's accordion movement in pipe. Fig. 4 shows the brief movement of an snake. When the snake moves a vertical pipe,

- (1) The snake bends the body like the accordion and holds the pipe by the whole body.
- (2) The snake stretches oneself to the moving direction. At this time, snake's lower part of the body continues the holding of the pipe.
- (3) The snake bends the upper part of the body and holds the pipe.
- (4) The snake extends the lower part of the body, and releases the holding of the pipe.
- (5) The snake hauled in the lower part of the body. Afterwards, the snake bends the lower part of the body and holds the pipe.

Thus, the snake can move in the vertical pipe.

MOVING PRINCIPLE OF THE ROBOT

The moving principle of the robot shown in Fig. 5. The moving principle of the robot referred to the accordion movement of the snake.

As for step 0, negative pressure is given to all bellows.

Step 1: The third somite is bent and it holds of pipe.

Step 2: Positive pressure is given to the first and second somite and somite are stretched.

Step 3: The first somite is bent and it holds of pipe.

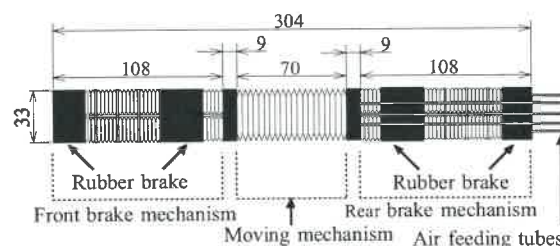


FIGURE 1. Structure of in-pipe mobile robot

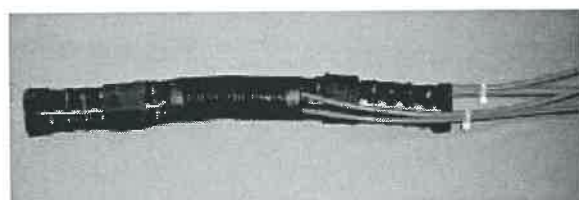


FIGURE 2. The photograph of in-pipe mobile robot

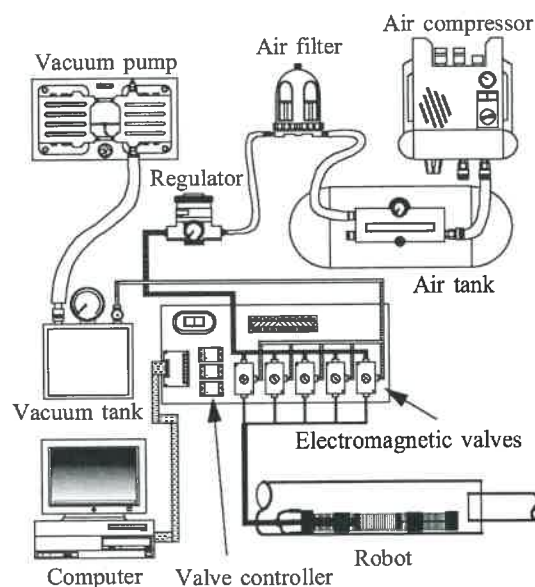


FIGURE 3. Experimental apparatus

Step 4: Negative pressure is given to the third somite and the holding of the pipe is released.

Step 5: Negative pressure is given to the second somite. The latter half of the robot advances in this step.

Step 6: The third somite is bent and it holds of pipe.

The robot can move by repeating these steps.

MOVING EXPERIMENT

Moving speed of the robot

The moving speed of the robot is measured. Fig. 6 shows the experiment result. The using pipes to experiment are 44 mm, 60 mm, 70mm, 79 and 90 mm in the inner diameter. The air pressures used to experiment are 50 kPa and -80 kPa.

The robot is able to move in these pipes and a combined pipe of a different diameter. The moving speed confirmed no relation to inner diameter of pipe and average moving speed was 25 mm/s.

Moving test in a T-junction

We fabricated a T-junction of pipe line for moving test shown in Fig. 7. The pipe has 79 mm inner diameter. The front brake mechanism of the robot can be decided the direction of the bending by controlling the supplied pressure. The fabricated mobile robot is confirmed to move in T-junction and was able to move in the desired direction.

Brake mechanism characteristics

We measured the friction force that became brake force of a in-pipe mobile robot. Five kinds of acrylic pipes of the inside diameter 44 mm, 60 mm, 70 mm, 79 mm, and 90 mm were used in the experiment. Internal pressure of two bellows that compose the brake mechanism was changes in the experiment. Internal pressure of an inside bellows was controlled to be constant -80 kPa and the pressure supplied to an outside bellows was changed with -50 kPa, -25 kPa, 0 kPa, + 25 kPa and + 50kPa. The frictional force that occurs for the inside diameter of the pipe in shown in Fig. 8.

The measured friction force was 6.8 N in the pipe of 44 mm in the inner diameter when pressure in an outside bellows was +50 kPa,

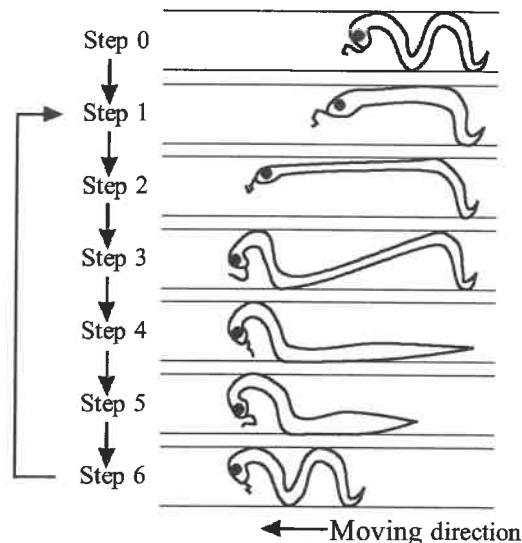


FIGURE 4. Accordion movement of a snake

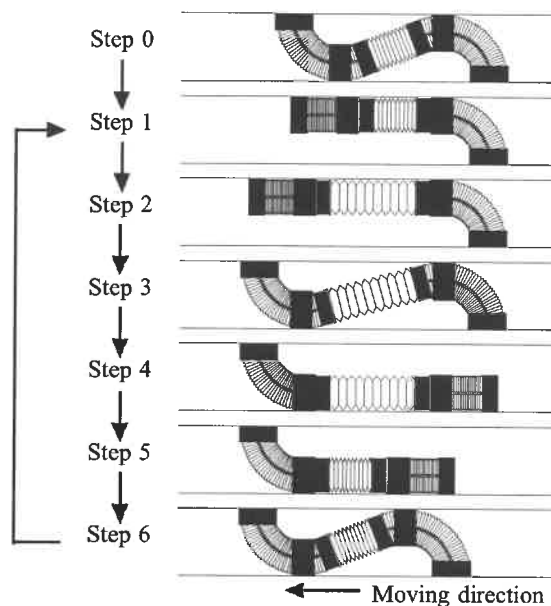


FIGURE 5. The moving principle of the robot

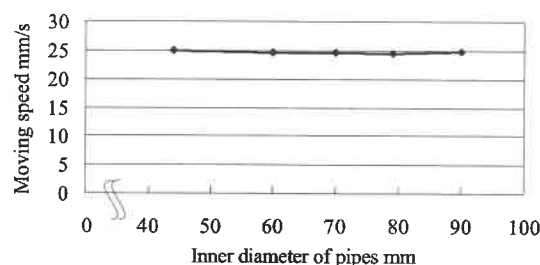


FIGURE 6. The moving speed characteristics

and 2.5 N in the tube of 90 mm in the inner diameter. Therefore, the friction force was confirmed to depend on the inner diameter of the pipe. When pressure in an outside bellows was -50 kPa, the measured friction force was 2.6 N in the pipe of 44 mm in the inner diameter. Therefore, it has been understood that the friction force depends also on the pressure in an outside bellows of brake mechanism.

The pressure supplied to an outside bellows was controlled and the normal force which was generated at the braking mechanism was measured. The normal force that occurs for the inside diameter of the pipe is shown in Fig. 9. When pressure in an outside bellows was +50 kPa, normal force was 1.9 N in inner diameter 90 mm. Moreover, it was 2.3 N in the pipe of 44 mm in the inner diameter. The normal force is larger in case of the pressure of +50 kPa than the case of the pressure of -50 kPa. Therefore, it has been understood that the generated normal force depends also on an internal differential pressure of two bellows of the brake mechanism.

CONCLUSIONS

We propose the robot which is imitate the snake of accordion movement. The fabricated mobile robot is confirmed to move in different diameter pipes and T-junction whose diameters are between 44 mm and 90 mm. Its speed was 25 mm/s.

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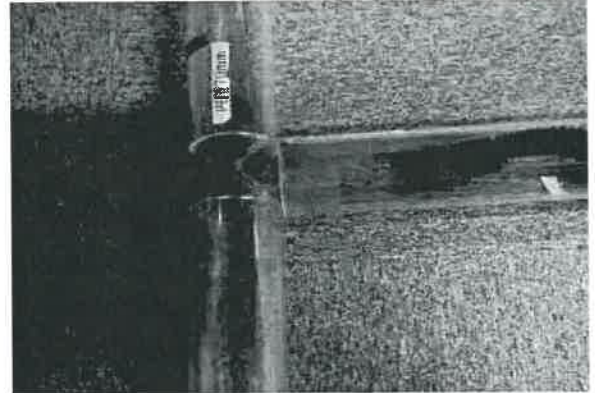


FIGURE 7. T-junction pipeline

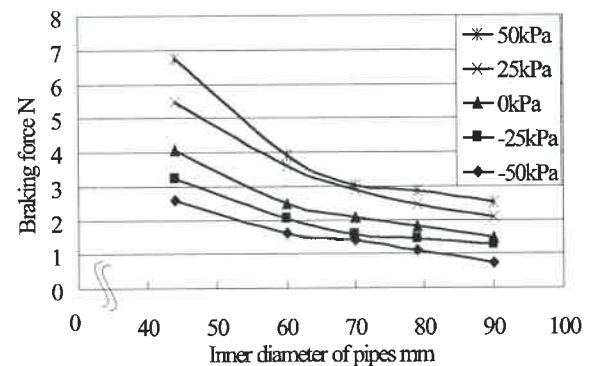


FIGURE 8. Relationship between diameter of pipe and braking force of robot

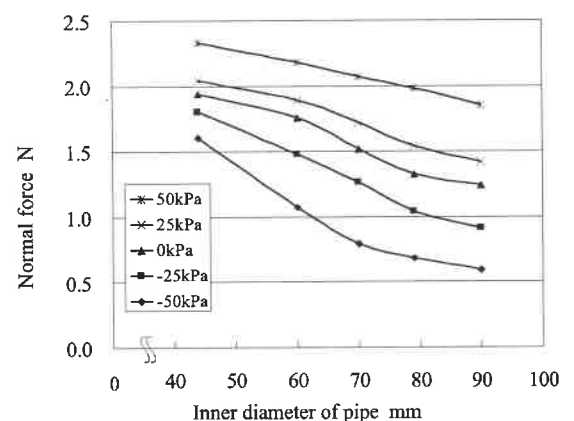


FIGURE 9. Relationship between pipe diameter and normal force

(2003), 564.

COMBINED MICRO MILLING AND LASER ABLATION PROCESSES IN A HYBRID MACHINING SYSTEM

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ABSTRACT

This paper introduces a machining center that enables laser ablation and micro milling processes sequentially without removing the workpiece from the workspace. Furthermore, an optical measurement module is integrated for in situ acquisition of the workpiece's geometry.

A mold insert manufactured by using the hybrid process chain demonstrates feasibility and advantages of the combined manufacturing technologies.

INTRODUCTION

Small and medium lot sizes of parts containing micro structures are needed to an increasing degree. They are used for example as micro dies and moulds for injection molding or micro punching. These parts can be manufactured economically by milling with tools of low nominal diameter. An alternative production technique that affords huge latitude concerning geometry features and machinable workpiece materials is laser ablation.

Here, a hybrid process chain is realized inside one machine tool, consisting of a milling process and a laser ablation process for the manufacturing of micro parts.

State of the art in die and mould making is a process sequence in which micro milling is used mainly to produce the electrodes for the die-sinking process. But in the meantime, reliable micro milling processes are possible by the use of tools with a low nominal diameter even for difficult to cut materials such as hardened tool steel. But in the tool diameter range less than 0.5 mm, milling of sharp corners in tool steel hardened up to a Rockwell hardness of 60 HRC or higher e.g. is still challenging. Tool deflection and tool wear have significant influence on the quality of the manufacturing process in this context. Beyond this, by principle, the sharpness of a corner is restricted by the tool diameter.

The use of state of the art picosecond laser beam sources for laser ablation offers the ability to overcome the geometrical restriction, but the removing rate of the laser cutting process in comparison to micro milling is smaller by several orders in magnitude.

Against this background, the basic intention of the introduced machine tool is to combine the advantages of both technologies – micro milling and laser ablation:

- to reduce the often large number of electrodes for Electrical- Discharge- Machining (EDM)- Processes,
- to overcome the geometrical restrictions caused by the geometry of the milling tool,
- to widen the range of machinable workpiece materials,
- to utilize a single clamping for the complete machining of complex micro parts,
- to make use of laser marking and
- to integrate quality estimation procedures in situ by using the optical gauge.

MACHINE TOOL SETUP

General Configuration

Figure 1 shows the hybrid machine tool. It is a portal construction, having a compactly designed cross table at the base. The vertical axis is placed at the crossbeam of the frame. The translational axes are equipped with synchronous linear drives. There are two motors for each axis arranged symmetrically and facing, leading to abrogation of the attraction forces of the motors. Five-axis-machining is made possible by a compact direct driven tilt/ swivel unit, that is integrated into the cross table.

The overall design concept allows for the need of highly dynamical movements of the machine tool axes with high jerks for both micro milling and laser machining. The frame is designed with a high static and dynamic stiffness as well as high eigenfrequencies.

The control hardware in the background of figure 1 includes the PC-based NC (numeric control) for milling, the control for the optical gauge including data acquisition as well as the laser process and beam source control.

The workspace is shown in the lower image of figure 1. The process modules are arranged radially on a revolver, which is placed at the z-carriage. By rotating this revolver, each module can come into effect.

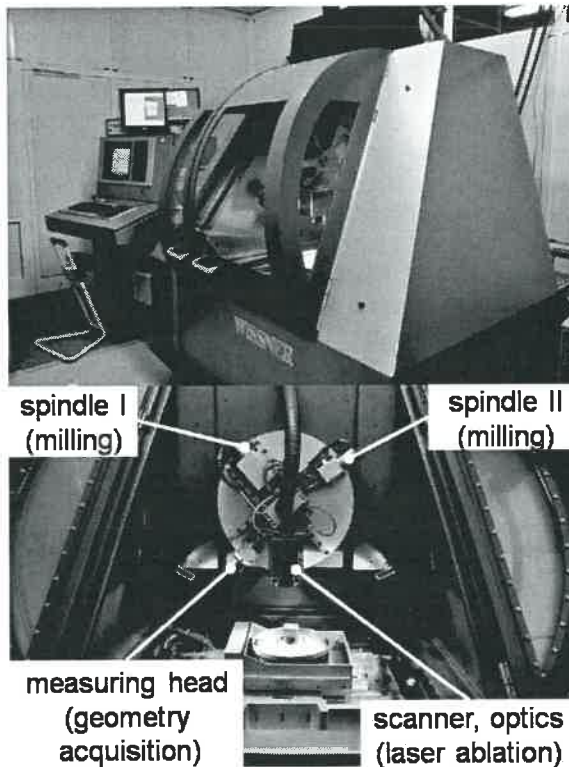


FIGURE 1. Machine tool for combined processes (top) and its workspace with modules (bottom)

For the use of the sensor for acquisition of the workpiece geometry, the machine tool axes realize the relative movement between workpiece and probe. For laser ablation, the z-axis of the machine tool is used, and the movement of the laser focus in the horizontal plane is realized by a scanner having mirrors driven by galvanometer motors.

The laser beam source rests on the frame's base, the beam is threaded into the scanner device from the back.

Micro Milling

The machine tool is equipped with two high frequency ball bearing spindles and automatic

tool change. The Tool paths are generated with a CAM system outside the machine tool control.

Laser Ablation

Laser ablation is realized using a picosecond laser source with pulse duration of less than 15 ps and with a pulse repetition rate up to 500 kHz. The process uses UV-light with wavelength 355 nm, mean power of 800 mW at maximum and peak power in the range of 650 MW.

The principle of the laser ablation process is to divide a 3d-model of the geometry to be removed into slices and to hatch the slices with the laser beam focus. This is automatically done by the laser CAM and control software installed on the machine tool control PC.

The scheme for generation of laser focus paths is illustrated in figure 2. The CAD geometry of the volume to be removed by the laser process is converted into the STL-data-format. This STL file is divided into the slices. The thickness of one slice corresponds to the material removal of one scanner path with constant machine tool axis z-position and lies in the region of 100 nm for finishing and a few micrometers for roughing, depending on material and laser parameters. The slices are hatched with a constant hatch distance.

The laser ablation process works relatively to the actual surface of the workpiece. Hence, every deviation between actual geometry and proposed geometry from CAD will be "imprinted" into the next slice. To make use of the laser machining in combination with milling therefore requires knowledge of the exact geometry after a single process step. E.g. if the intention of a laser ablation process after a milling process is to remove burr, the CAD data has to contain the exact geometry of the burr.

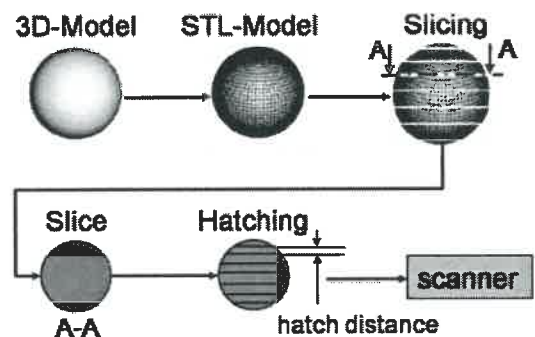


FIGURE 2. Laser Ablation on the basis of 3D-CAD data

Workpiece Geometry Measurement

The CAD data of the actual workpiece geometry is generated by use of measurement data from an integrated module. The module consists of an optical probe, based on the chromatic aberration measurement principle. The probe has a resolution of 30 nm and an accuracy better 0.5 μm . The measurement range is 1 mm. The acquisition of the distance between workpiece surface and optical probe is done with 14 kHz. Each measurement datum is stored together with the position values of all encoders of the machine tool axis allowing an exact reconstruction of the measurements for a point cloud. The sensor path can either be generated by the CAM System, thereby using the sensor as a tool. Alternatively the sensor path can be generated by a CNC-plugin for data acquisition line-by-line. The CNC-plugin also allows a fast visualization of the workpiece's surface based on the acquired point cloud.

Hybrid Process Chain

The hybrid process chain consists of milling and laser ablation processes in any order with geometry measurement in-between. The hybrid process chain, demonstrated here consists of the process steps shown in figure 3. The initial point is the CAD model of a mould insert. By using a CAM system for tool path generation, the cavity is milled firstly. After the milling process, the sensor is moving along the workpiece in constant z-height line by line. The next step is to generate the geometry to be removed by the laser. The laser ablation process is then used for finishing the mould insert.

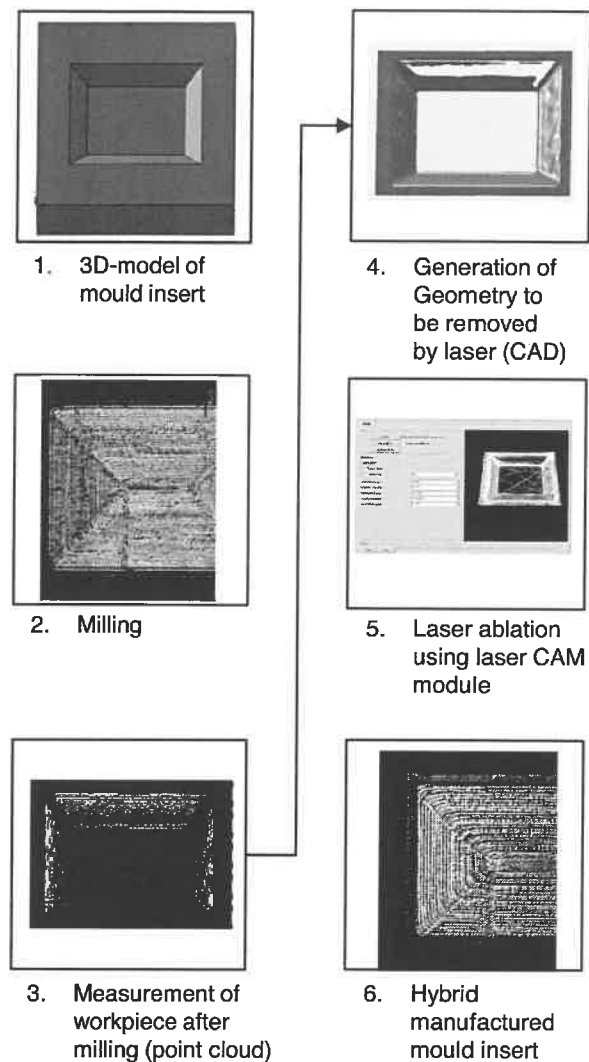


FIGURE 3. Process steps of hybrid manufacturing process

RESULTS

A mould insert with a footprint of 4 mm by 5 mm and a depth of 0.7 mm had been manufactured using the hybrid process chain. The insert is made of tool steel that has been hardened up to a Rockwell hardness of 62 HRC. The cavity to be machined is a rectangle with edges of 45°, as shown in figure 3, step 1.

Firstly, the cavity is milled with cemented carbide end mills and ball end mills with a nominal diameter of 0.5 mm. After milling, the surface geometry of the workpiece is acquired by using the chromatic aberration sensor. The point cloud's quality and its high density allows the generation of a surface CAD-model of the

actual geometry of the workpiece's surface directly without using filters for preparation of the measurement data for surface approximation. This is done on a separate PC because of the huge amount of data. The surface model can be generated using triangulation algorithms or surface approximation. Here, the approximation with a surface of 100 by 100 segments with order 3 was feasible.

A standard CAD system was used to generate a 3D-model of the volume difference between intended geometry and the measurement result. This CAD Model is exported as a STL-file and imported into the laser control software. The entire workflow including the point cloud is done in one coordinate system.

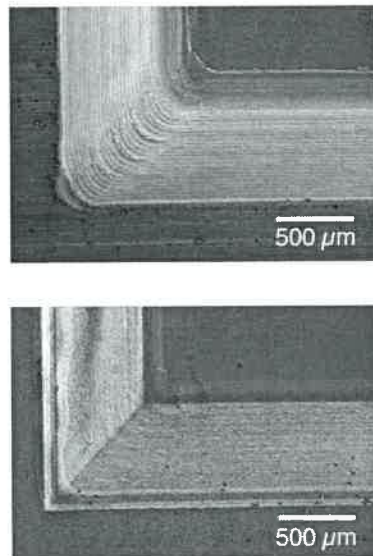


FIGURE 4. SEM-Images showing a corner after milling (top) and after hybrid machining (milling, measuring, laser ablation) (bottom)

The laser ablation process parameters have been

- hatch distance/ slice thickness according figure 2: $4.00\ \mu\text{m}$ / $4.37\ \mu\text{m}$,
- Mean laser power: 640 mW,
- Repetition rate: 500 kHz and
- Laser feed rate: 800 mm/s.

The SEM images in figure 4 show a corner after the milling process (top) and a corner after laser ablation on the basis of the acquired workpiece geometry after milling (bottom). It can be ascertained, that laser ablation is able to sharpen the corner and to remove the stair-like structure of the milling path.

The SEM-images in figure 5 show the transition of surfaces reworked with the laser and surfaces only generated by milling.

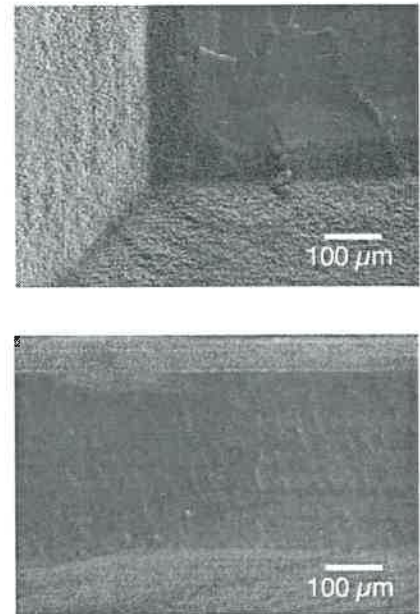


FIGURE 5. SEM-Images: Details of corner at the ground (top), and at an edge (bottom)

CONCLUSION AND OUTLOOK

The introduced machine tool eliminates losses in position accuracy of the workpiece caused by removing the workpiece from the machine tool and fixing, when using separate machine tools for milling and laser machining. The concept combines furthermore the (in comparison to laser ablation) high removal rate of the micro milling process with the (regarding the achievable geometry features) flexibility of the laser ablation process. This facilitates economical micro machining of difficult to cut materials as well as quality improvement of the machined part by immediate rework through integrated measurement.

Further investigations will be enforced regarding protection of optical/laser components while milling, regarding shortening of time needed for generating the CAD model from measurement data and regarding automation of generating the CAD model from measurement data.

ACKNOWLEDGEMENT

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MAGNETIC FIELD EFFECTS ON THE SECONDARY BEAT FREQUENCY PROFILE FOR THREE MODE HENE LASER STABILIZATION

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INTRODUCTION

In displacement interferometry applications, the laser wavelength must be stabilized to ensure the proper phase to displacement conversion factor. This is especially important when the measurement arm path length is long, because the uncertainty from wavelength fluctuations is linearly proportional to the length travelled [1].

Typically, HeNe lasers are frequency stabilized via a longitudinal Zeeman split or via a two mode intensity split in the commercial heterodyne interferometers. In applications where the stage velocity is slow, however, the high megahertz split frequency that result from these methods is unnecessary. Additionally, the heterodyne laser source using these methods have an optical power less than 1 mW and cause frequency mixing, which contributes to periodic nonlinearity in the measurement.

In this research, a three mode laser is stabilized using a technique which has been shown to have stabilities on the order of 10^{10} [2,3]. In short, using a three mode laser as the source, the two side modes can be blocked because they are orthogonally polarized from the central mode. The usable optical power from the central mode can be more than 2 mW. For heterodyne laser interferometers, the central mode can be split equally and passed through two separate acousto-optic modulators to obtain two spatially separated beams with a frequency difference of 10 kHz to reduce periodic nonlinearities.

Additionally, this research includes an investigation of transverse magnetic field effects on the three mode laser stabilization. Because the central mode of a randomly polarized three mode laser has an alternating polarization state, an additional feedback loop is required to ensure a correct polarization state. This work determines if applying a magnetic field relative to the nominal laser polarization would force a specific single central mode polarization and thus removing the additional control loop. Further research was done to determine the effects on frequency stability and output power.

THREE MODE LASER STABILIZATION

A three longitudinal mode Helium Neon laser was stabilized using the secondary beat frequency. Figure 1 shows a schematic of the three laser modes. The first outer mode, the center mode, and the second outer mode are F_{01} , F_c , and F_{02} , respectively. In the absolute frequency domain $F_{01} < F_c < F_{02}$, and F_c is orthogonally polarized from F_{01} and F_{02} .

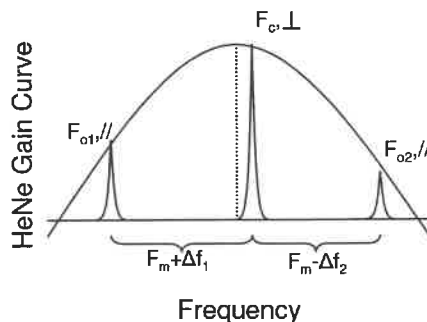


Figure 1: Schematic of the mode spacing in a three mode HeNe laser. Due to non-symmetry, and frequency pulling effects, the modes are not equally spaced.

The nominal frequency difference between the modes, F_m , is $c/2L$ where c is the speed of light, and L is the length of the laser tube. In the laser used in this experiment, the nominal mode spacing was approximately 688 MHz, meaning the laser operates in both two and three mode states.

Due to non-symmetry in the laser mirror coatings and frequency pulling effects, the differences between the two modes are not the same, and the central mode does not line up exactly in the center of the helium neon gain spectrum, which leads to frequency differences Δf_1 and Δf_2 .

By placing a polarizer (p) at the laser's output, the three modes interfere and several beat signals are created. Two beat signals at approximately 688 MHz come from the differences between the outer modes and the central mode. The useful beat signal for

frequency stabilization is the secondary beat, f_{2nd} , given by

$$f_{2nd} = F_m + \Delta f_1 - F_m + \Delta f_2, \quad (2)$$

or

$$f_{2nd} = \Delta f_1 + \Delta f_2. \quad (3)$$

The secondary beat is on the order of a few hundred kilohertz, which is detectable by a normal photodiode receiver (PD_L), rather than a more complex avalanche photodiode. This secondary beat is then the controller input and a thermo-electric cooler (TEC) stabilizes the laser tube's length.

ABSOLUTE FREQUENCY STABILITY

After the laser is stabilized, the central mode is isolated using a Glan Thompson polarizer (GTP) and Fabry-Perot optical spectrum analyzer (FP OSA). This central mode is then mixed with a commercial iodine stabilized laser using a beam splitter. A half wave plate (hwp) rotates the iodine laser's polarization to match the 3-mode laser's central mode. An avalanche photodiode (APD) measures the interference signal. Figure 2 shows this optical configuration.

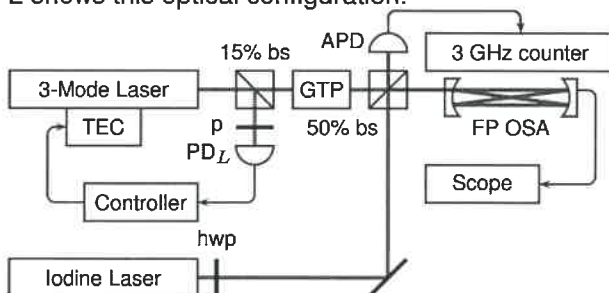


Figure 2: Optical configuration for measuring the frequency stability of the central mode of the three mode laser relative to an iodine stabilized laser. See text for abbreviations.

In this configuration, a 3 GHz counter measures the APD signal. Both the secondary beat frequency and absolute frequency are measured and fluctuations in the secondary beat frequency are clearly visible and proportional to the absolute frequency fluctuations. After correcting for these fluctuations with a simple linear fit, the fractional frequency stability of the three mode laser is better than 5 parts in 10^{10} , at 1 s integration time. The iodine stabilized laser (Model 100, Winters) has a frequency stability of 2.5 parts in 10^{11} at a 1 s integration time, which shows that the fluctuations in the beat between these two are coming largely from the three mode laser. The Allan variance of the three mode laser frequency stability is shown in Figure

3, showing the stability increases with integration time.

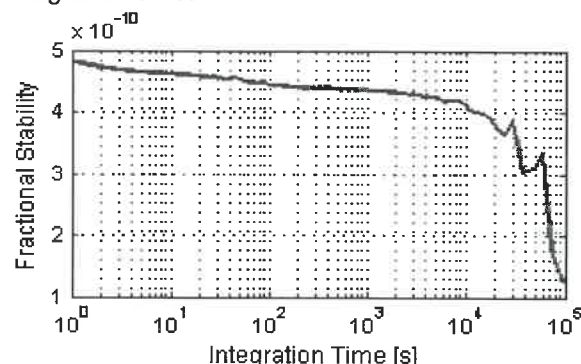


Figure 3: Allan variance of the central mode frequency stability, stabilized by the secondary beat frequency.

MAGNETIC FIELD EFFECT HYPOTHESIS

The achieved frequency stability (using the secondary beat frequency) is more than adequate for the initially intended application. However, this laser suffers from a random polarization of the central mode, i.e. the laser could stabilize in either a parallel (S) or perpendicular (P) states. This limits its uses in polarization sensitive interferometers. It is possible to measure the intensities of the two states using a polarizing beam splitter and only stabilize the laser when a specific polarization has a higher intensity. But, this complicates the optical setup by adding 2 beam splitters, 2 detectors, and data acquisition components.

Alternatively, applying a transverse magnetic field to the laser tube may override the laser's initial polarization because the laser's standing wave electric field has an associated magnetic field.

In this work, we test if aligning a transverse magnetic field with the right field strength will cause the central mode to always be a single polarization. Additionally, this might increase the output power of the central mode due to amplification from the magnetic field, which is proportional to the electric field amplitude.

MAGNETIC FIELD OPTICAL LAYOUT

Figure 4 shows the setup used to test the magnetic field effects. The laser beam first passes through an initial half wave plate. Then, this beam is split using a beam sampler, where one portion passes through a polarizer and is detected using a photodiode. The remaining main beam is split equally using a beam splitter. One half passes through a polarizing beam splitter and to two detectors to measure the DC

power of the polarization states. The other beam passes through a GTP and then detected using another photodiode. Both the AC and DC portions of this signal are measured to assess frequency mixing and output power differences.

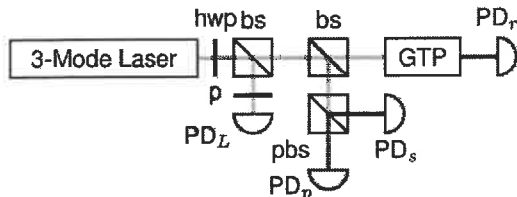


Figure 4: Optical configuration for the magnetic field effect experiments.

The HWP is used to align the initial polarization state of the laser such that the main mode passes through the GTP and the PBS, and the two outer modes are reflected by the PBS and blocked by the GTP. The DC level from the photodiode PD_r measures fluctuations in output power and the AC portion determines if there is a rotation of the nominal laser polarization from the magnetic field.

MAGNETIC FIELD RESULTS

For the experiments, the transverse magnetic field angle was rotated through 180° and the polarizer angle was rotated 90° , both in increments of 15° . The laser was allowed to warm-up to steady state where the effect of multiple mode hops was observed. This observation showed a clear distinction between two and three mode operation.

The profile of the secondary beat during three mode operation was recorded based on the main polarization state of the central mode. This can be seen in Figure 5 where the profiles for a specific polarizer angle and magnetic field angle are shown over a full mode sweep.

When there is no magnetic field applied, the central mode during three mode operation alternates between S- and P-states. However, when the magnetic field is applied at 60° to 120° , there is only a P-state from the central mode. Conversely, the central mode is S-state when the magnetic field was between 0° - 30° and 135° - 165° . In the other magnetic field orientations (blank in Figure 5), no usable three mode sweep signal were detected.

This means the applied magnetic field can force the three mode laser to stop alternating polarization states, allowing a simpler optical system in practical applications. As shown in Figure 5, a 60° applied magnetic field produced the most consistent and linear portion for the three mode sweep. It is possible to stabilize the laser with the magnetic field at this orientation; however, no comparison was made between the frequency stability against the Iodine reference laser.

Additionally, with the applied magnetic field, a single polarization state could be forced from the central mode. However, no polarization rotation was observed. Because of the differing shapes of the three mode profiles, no conclusive observations of findings from this portion of the research were made.

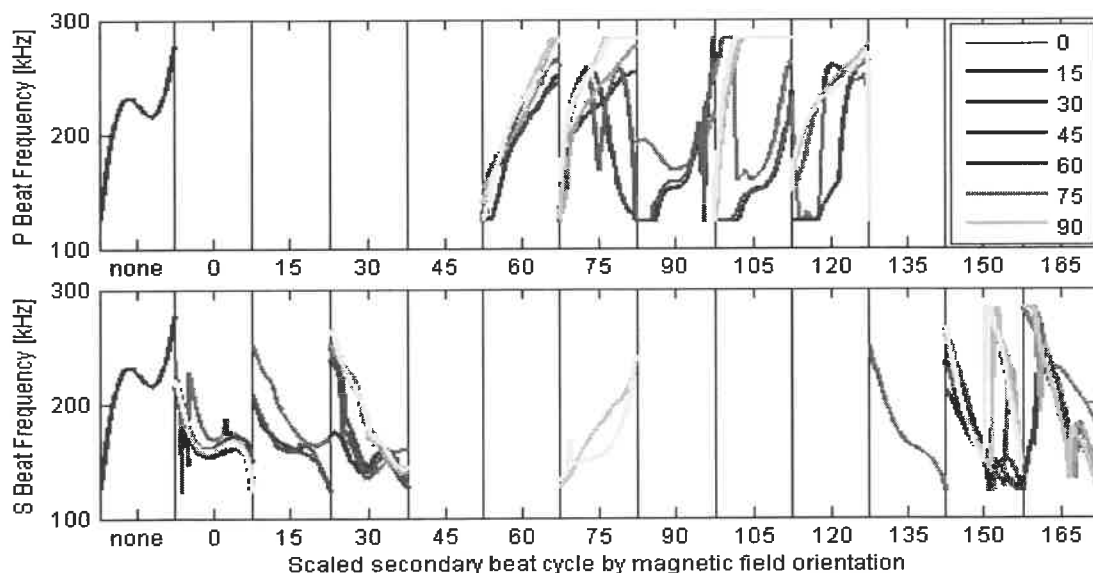


Figure 5: Three mode profiles for varying magnetic fields based on the main polarization state from the central mode. The blank sections showed no usable secondary beat signal.

ADDITIONAL THREE MODE LASER TESTS

The magnetic field tests were largely inconclusive because an overall pattern based on the magnetic field could not be determined. Therefore, a more fundamental understanding of the three mode laser without the applied magnetic field was needed.

A non-magnetized three mode laser was used to verify so-called "super heterodyne" techniques which double the optical resolution without doubling the optical path length [4]. As shown by Yokoyama, *et. al.* [4], the secondary beat signal was detected using an avalanche photodiode instead of the low speed detector used by Suh, *et. al.* [2] and Yeom, *et. al.* [3]. Additionally when the polarizer in Figure 2 was removed, there should be no signal present on PD_L because the detector bandwidth was much lower than the mode spacing. However, after filtering and amplifying, a signal at the same frequency as the secondary beat was observed. Additionally, this signal was successfully used to stabilize the laser's absolute frequency, which was unexpected. To fully understand the three mode stabilization technique, the configuration shown in Figure 6 was used.

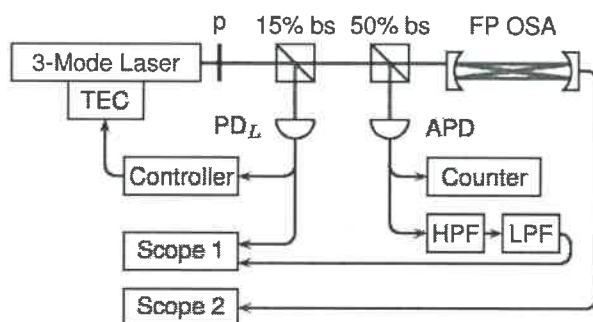


Figure 6: Optical configuration for determining the performance of the three mode laser.

In this configuration, the polarizer is common to the PD_L, the APD, and the Fabry-Perot optical spectrum analyzer (FP OSA), which differs from our original setup and previous research. When the polarizer is at 0°, only the center mode should be present on the FP OSA and no secondary beat frequency should be detectable. However, this is not the case, which can be seen in Figure 7. In this figure, the signal from the FP OSA was captured, which distinctly shows a central mode with the polarizer at 0°, all three modes at 45°, and the two outer modes at 90°. At 0° and 90°, the peak-to-peak signal from PD_L was ~0.45 V. At 45°, there was no detectable secondary beat signal, which is

different than what has been previously quoted by Suh, *et. al.* [2] and Yeom, *et. al.* [3].

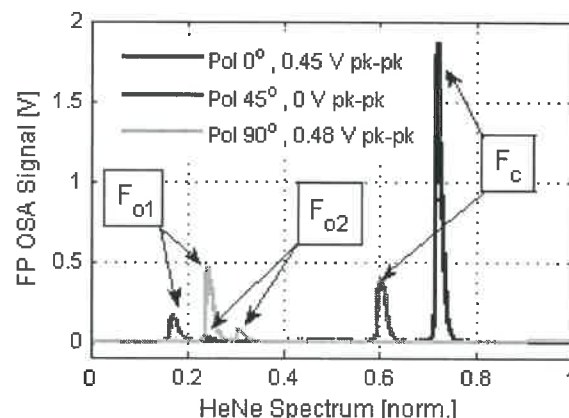


Figure 7: Mode sweeps with polarizer at 0°, 45°, and 90°. Secondary beat signals were detected at 0° and 90° but not 45°.

CONCLUSIONS

We successfully demonstrated absolute frequency stabilization of a three mode laser based on the secondary beat signal. The frequency stability after correcting for correlated fluctuations in the secondary beat signal was better than 5 parts in 10¹⁰ at an integration time of one second.

Additionally, it is possible to use a transverse magnetic field to change the profile of the secondary beat signal and force the laser to have a specific polarization of the central mode. However, given the discrepancies about the actual three mode operation of the laser from Figures 6 and 7, more research is needed to validate using the transverse magnetic field effectively.

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IMPROVED SAUNDERS METHOD USING LATERAL SHEARING INTERFEROGRAMS OBTAINED BY SEVERAL DIFFERENT SHEAR

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1. INTRODUCTION

Mirrors for high performance optical systems are manufactured with ultra-precision lathes. Measurements on this machine are necessary for the precise processing of mirror shape. A lateral shearing interferometer is a kind of common type of path interferometer, which is little affected by mechanical vibrations and air turbulence. In the lateral shearing interferometer, interferograms are produced by the optical path difference between a wave front under testing and a laterally sheared wave front. Therefore, the interferograms produced by a lateral shearing interferometer show a contour map of the difference of the wave front under test in the direction of the shear. The contour map must be analyzed to reconstruct the wave front under testing, because the contour map does not directly show the wave front. Many analysis methods for this reconstruction process have been proposed for lateral shearing interferograms. The Saunders method [1-2]

reconstructs the wave front rapidly, via a summation of the differences in the direction of the shear. However, the method has low spatial resolution, because the wave front is reconstructed only at shear intervals along the direction of the shear. An improved Saunders method [3] has also been proposed for high resolution measurement. This improved method analyzes the interferograms produced by shearing the wave front under test in many directions [4]. In the improved method, a large number of interferograms is required for reconstruction of the wave front, and the analysis process is complex. In this paper, we propose another improved method based on the original Saunders method. In the proposed method, interferograms are acquired for two different conditions for amount of shear. Each of these interferograms is analyzed by the original Saunders method, and the analysis results are connected without approximation using a simple method [5]. In this paper, the operation of the

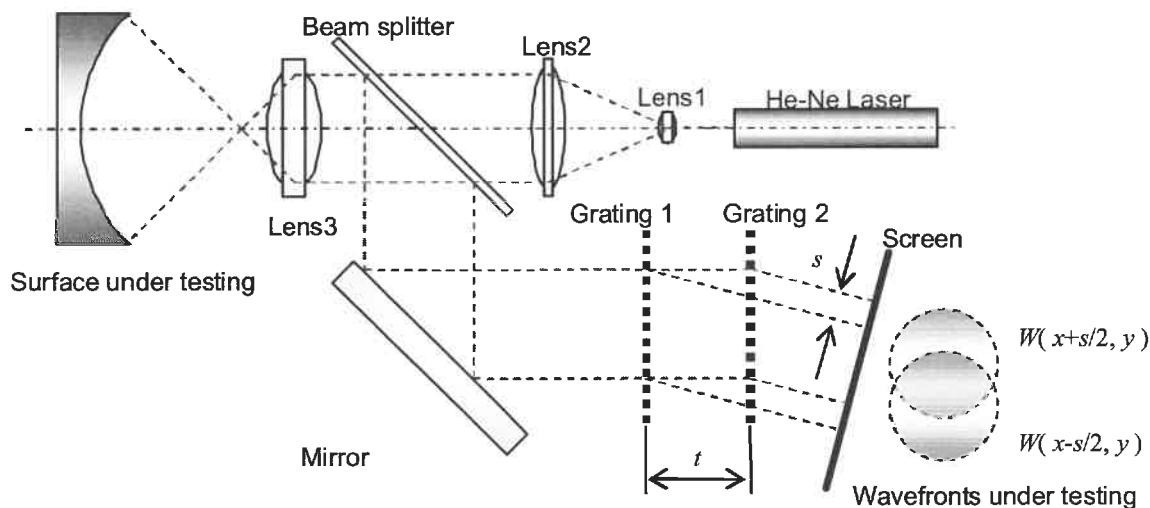


FIGURE 1. Lateral shearing interferometer

proposed method, the shear conditions, and an optical system to vary the amount of shear are described in detail. The validity of the proposed method is then confirmed by experiments, in which a concave mirror with shape errors is measured using a lateral shearing interferometer and the resulting interferograms are analyzed via the proposed method. The results obtained using the proposed method are compared with those obtained using a conventional method.

2. LATERAL SHEARING INTERFEROMETER

Figure 1 shows a schematic drawing of a lateral shearing interferometer with two gratings. In the interferometer, a laser beam is collimated by lenses 1 and 2. The collimated beam is refracted by lens 3 and reflects off of a surface under testing. When the surface has errors, the wave front of the reflected beam is deformed. The wave front is divided by two gratings. The resulting two wave fronts are laterally sheared with respect to each other. amount of shear, s is expressed as:

$$s = \frac{t \sin(2\theta)}{2}, \quad (1)$$

$$\theta = \sin^{-1}\left(\frac{\lambda}{p}\right), \quad (2)$$

where t is the interval between the two gratings, θ is the diffraction angle of the beam at the gratings, λ is wavelength of the beam, and p is the pitch of the gratings. A lateral shearing interferogram is obtained from the area where the two wave fronts overlap. The phase of an interferogram changes in proportion to variations in the distance of the grating perpendicular to optical axis. Because the interferogram shows the difference between the wave front under testing and the sheared wave front, the interferogram must be analyzed to evaluate the wave front shape.

3. IMPROVED SAUNDERS METHOD

3.1 PRINCIPLE

The Saunders method evaluates the wave front via a summation of the lateral shearing interferogram at intervals of the amount of shear, as shown in Figure 2(a). When the

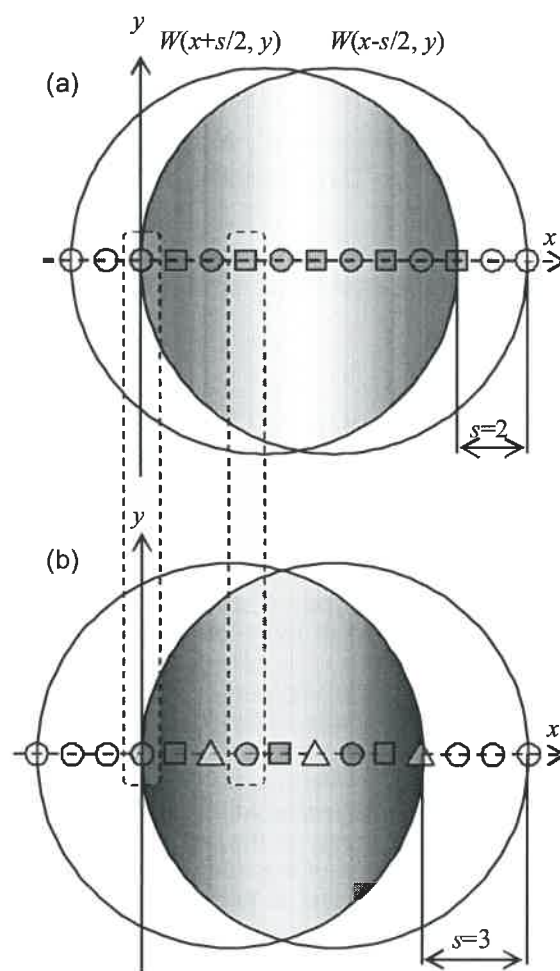


FIGURE 2. Relationship between the wave fronts and amount of shear

amount of shear is $s = 2$, the wave front shape is obtained at both the figure points represented by red circles and blue squares, at intervals of the amount of shear. However, these two groups are unrelated.

To connect the points of the two groups, a second interferogram with a different amount of shear (e.g., $s = 3$) is analyzed by the Saunders method, as shown in Figure 2(b). Figures 2 (a) and (b) are then compared at the points surrounded by dotted vertical rectangles. At these points, connecting information is obtained to compare the shape analyzed by shear amounts $s = 2$ and $s = 3$. The shape can then be defined exactly, and analyzed without approximation via the connecting information.

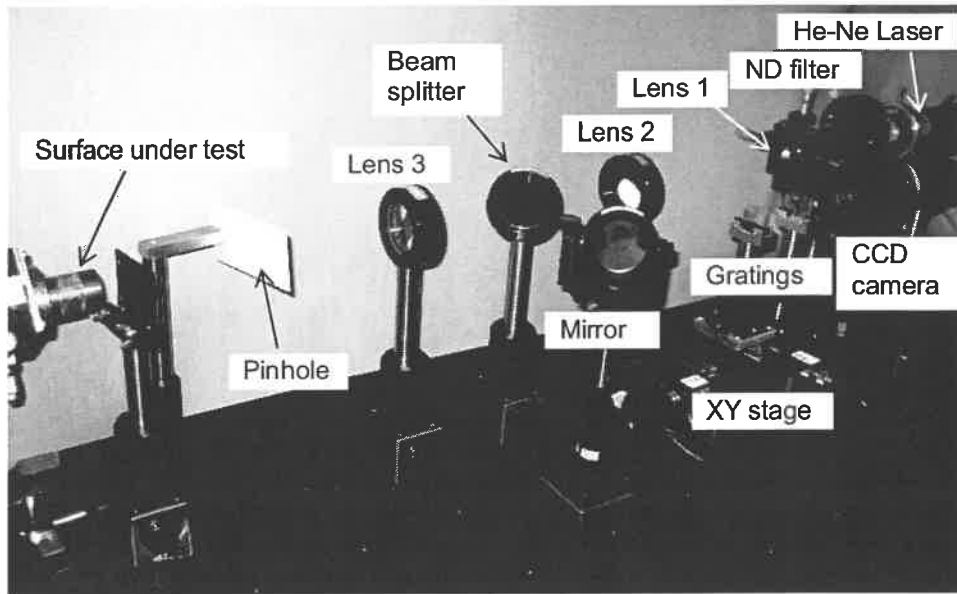


FIGURE 3. Photograph of lateral shearing interferometer

3. 2 NECESSARY SHEAR CONDITIONS

Wave front reconstruction based on an analysis from shear amounts, s_1 and s_2 requires the satisfaction of three conditions. Condition 1 is expressed as:

$$s_2 \neq as_1 \quad (s_1 < s_2), \quad (3)$$

where a is a natural number not equal zero. Condition 2 is that s_1 and s_2 have no common factor. Condition 3 is denoted by:

$$\frac{L}{d} = \left(\frac{s_1}{d} - 1 \right) \frac{s_2}{d}, \quad (4)$$

where L is the necessary interferogram range and d is pixel size.

4. EXPERIMENTS

To investigate the validity of the proposed method, a concave mirror with shape errors was measured using the lateral shearing interferometer shown in Figure 3, and the resulting interferograms, shown in Figure 4 were analyzed. Figure 5 shows the phase of the interferograms obtained by the 4 step phase shift method. Figure 6 shows the shape of the wave front, as analyzed by (a) the proposed method and (b) the conventional method.

5. CONCLUSIONS

A conceptually new method of analysis for the lateral shearing interferometer is proposed. Results obtained by the proposed method are compared with those obtained by the conventional method, showing good qualitative agreement between analysis results obtained from the two methods.

6. ACKNOWLEDGMENT

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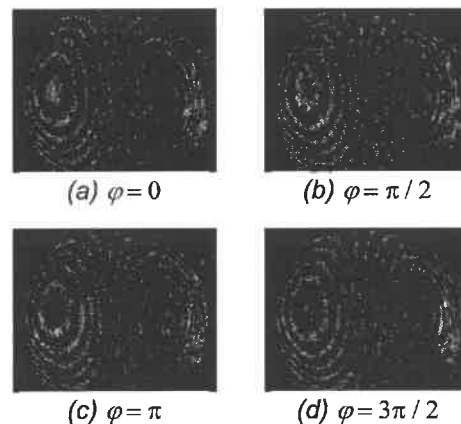
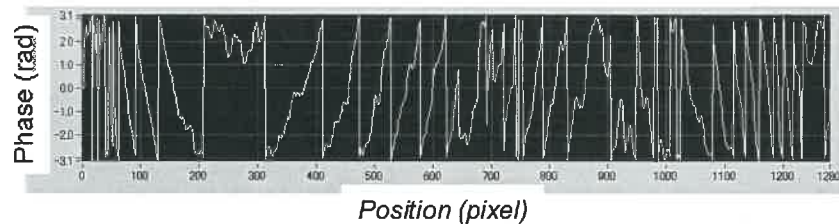
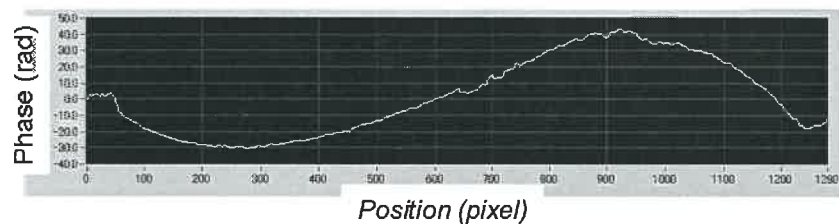


FIGURE 4. Interferograms obtained by a lateral shearing interferometer



(a) Wrapped phase distribution

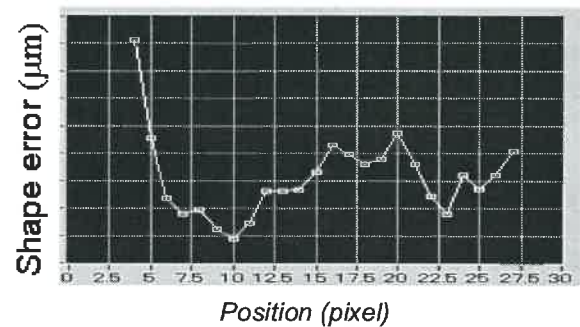


(b) Unwrapped phase distribution

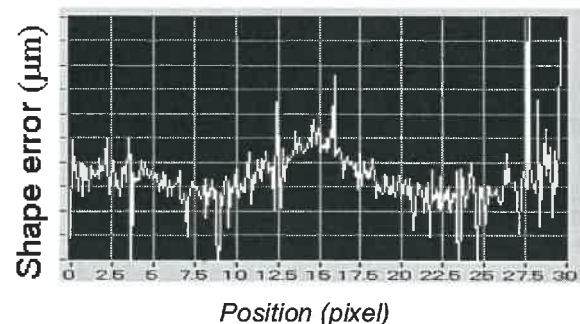
FIGURE 5. Phase distribution of the Interferograms obtained by a lateral shearing interferometer

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(a) Proposed method



(b) Conventional method

FIGURE 6. Results obtained by the proposed method and the conventional method

SIZE REDUCTION OF ACOUSTO-OPTIC MODULATOR-BASED HETERODYNE DISPLACEMENT MEASURING INTERFEROMETER (AOM DMI)

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INTRODUCTION

Displacement measuring interferometry has been extensively used in situations that require accurate displacement measurements, such as position feedback for lithographic stepper stages in semi-conductor fabrication, precision cutting machines, and coordinate measuring machines. A common configuration in these applications is the Michelson-type heterodyne displacement measuring interferometer where two slightly different frequencies are used and phase is measured to determine displacement.

Periodic error, a cyclical oscillation about the true displacement, can limit the achievable accuracy in these systems. In previous work [1], a new acousto-optic modulator-based heterodyne displacement measuring interferometer (AOM DMI) was developed to eliminate periodic error by avoiding frequency mixing. The purpose of this study is to reduce the footprint of the initial AOM DMI setup. A limitation of the AOM DMI is that the diffraction angle for the two beams exiting the acousto-optic modulators is small (7 mrad). This causes a large system footprint because significant lengths are required to obtain sufficient separation between the diffracted beams to be able to direct them along the measurement and reference paths. In the new design, a right angle prism with a small hole is introduced to modify the beam paths and reduce the interferometer size. Test results for the new setup are presented and the size comparison between the original and the new setup is provided.

PERIODIC ERROR BACKGROUND

Imperfect separation of the two polarization-coded frequencies at polarization dependent optics has been shown to produce first and second order periodic errors, or errors of one and two cycles per wavelength of optical path change, respectively. In a perfect system, a single frequency would travel to the fixed target, while a second, single frequency traveled to the moving target. Interference of

the combined signals would yield a perfectly sinusoidal trace with phase that varied, relative to a reference phase signal, in response to motion of the moving target. However, the inherent frequency leakage in actual implementations produces an interference signal which is not purely sinusoidal and leads to periodic error in the measured displacement.

An early investigation of periodic error in heterodyne interferometry was performed by Quenelle [2]. Subsequent areas of research have included efforts to measure periodic error using frequency domain analysis [3], analytical modeling techniques [4], and reduction of periodic error [5-7], for example.

Schmitz and Beckwith [8] described the potential periodic error contributors using a Frequency-Path model, which identified all possible paths for each light frequency between the source and detector and predicted the number of interference terms that may be detected at the photodetector output. It was demonstrated that 10 distinct interference terms exist in a fully leaking single pass, polarization-coded heterodyne interferometer. These interference terms may be grouped into four categories based on optical path change: 1) *optical power* which represents the dc power of each term and is independent of optical path changes; 2) *ac reference* terms with phase that varies by one cycle per wavelength of optical path change; 3) *dc interference* terms, which are Doppler up-shifted from zero frequency during target motion; and 4) *ac interference* terms which produce a time harmonic variation in the detector current at the beat frequency (f_b) and are Doppler shifted (f_d) up or down during target motion, depending on direction. The frequency content for a fully leaking heterodyne interferometer during constant velocity motion is depicted in Fig. 1.

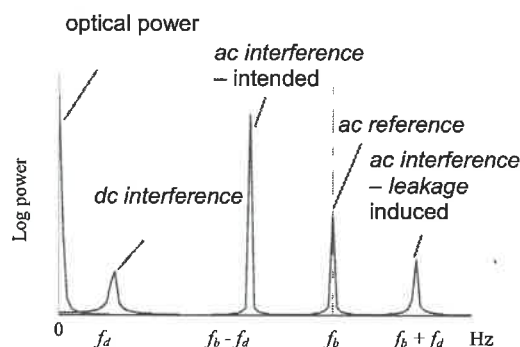


FIGURE 1. Frequency spectrum for constant velocity motion in fully leaking heterodyne interferometer.

EXPERIMENTAL SETUP

A photograph and schematic diagram of the reduced size AOM DMI setup are provided in Fig. 2.

In Fig. 2(b), a single frequency stabilized He-Ne laser is used as the light source. The output beam of the He-Ne laser is collimated and passes through the first acousto-optic modulator (AOM1), oriented at the Bragg angle, which produces the 0th order and 1st order (diffracted) beams. The right angle prism with hole is inserted at the shortest distance from AOM1 where the 0th and 1st order beams are visually separated. The 1st order beam is directed toward the fixed retroreflector for the measurement signal and one of the two (fixed) retroreflectors for the reference signal (Ref. RR1) by reflecting at the prism surface, while the 0th order beam continues toward the second acousto-optic modulator (AOM2) by passing beside the prism. The 1st order (diffracted) beam from AOM2 is directed toward the moving target retroreflector and second retroreflector for the reference signal (Ref. RR2). The retroreflected beam from

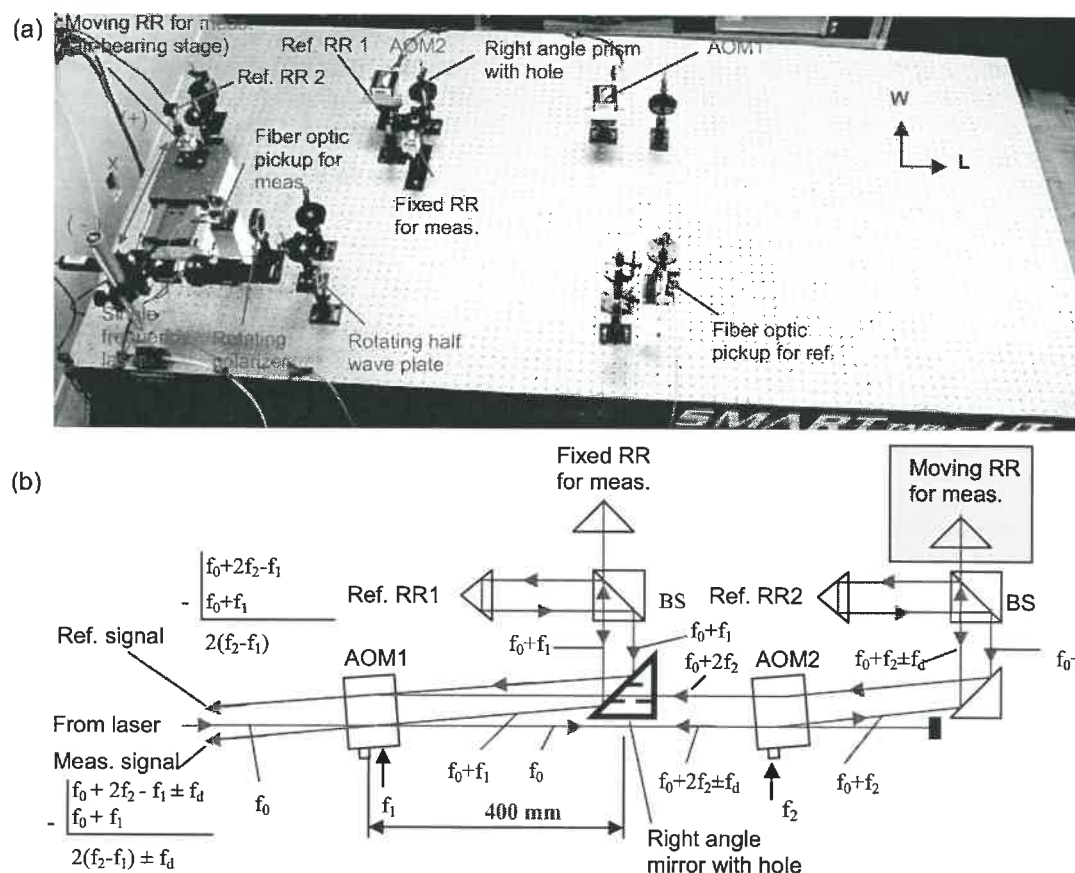


FIGURE 2. (a) Size reduced setup using a right angle prism with a hole. (b) Schematic of the new setup showing the beam separation and combination at each AOM and beam path through the right angle prism hole. (RR is used to abbreviate retroreflector.)

Ref. RR2 returns to AOM2 and passes through the hole of the right angle prism to recombine and interfere at AOM1 with the reflected beam from the Ref. RR1. The frequency of the reference signal is two times the frequency difference between the two AOMs driving frequencies. The retroreflected beams from both fixed and moving retroreflectors are also recombined at AOM1 to serve as the measurement signal. The frequency of the measurement signal is the same as the reference signal for no target motion, but up or down-shifted by the Doppler frequency during target motion (depending on direction). Finally, both the measurement and reference signals are passed to the photodetectors using multi-mode fiber optics. With the new setup (1400 mm (L) x 620 mm (W)), approximately a 65% area reduction from the original setup (2240 mm (L) x 1100 mm (W)) is achieved.

As noted, in the new setup design, a right angle prism with a hole is used to reduce the size relative to the initial setup. Figure 3 shows the prism with dimensions of 15 mm x 15 mm x 21.2 mm (A x B x C). The prism material is BK7 glass with an enhanced aluminum coating on the hypotenuse. To fabricate the required hole through the prism center, an electroplated diamond core drill was used. Figures 4(a) and 4(b) show the diamond core drill and its surface, respectively. The diameter of the beam passing through the prism is approximately 2 mm so that a 4.8 mm (3/16") hole provided sufficient clearance. Figure 5 shows the drilling operation setup, where the right angle prism was submerged in the water to reduce heating. The drilling was operated at a feedrate of 0.23 mm/min. The prism after drilling is shown in Fig. 6.

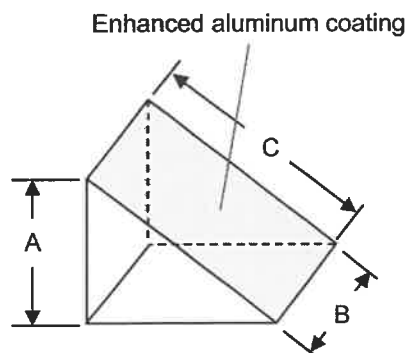


FIGURE 3. Technical drawing for a right angle prism.

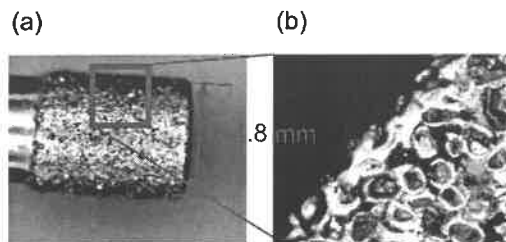


FIGURE 4. (a) An electroplated diamond core drill with the outer diameter of 4.8 mm. (b) Diamonds coating on the surface of the drill.

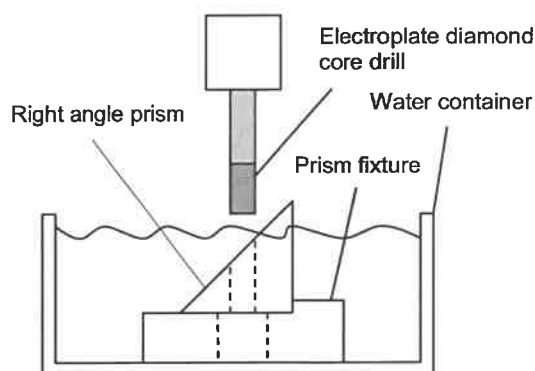


FIGURE 5. Schematic of drilling operation for a right angle prism using a diamond core drill in the submerged (water) environment.

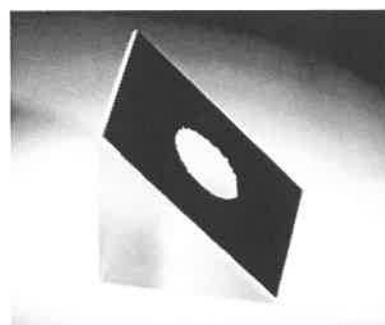


FIGURE 6. The right angle prism with hole at the center, fabricated using a diamond core drill.

EXPERIMENTAL RESULTS

In this section, frequency content of the measurement signal collected using an analog spectrum analyzer for the new setup is presented. Since the beat frequency, or frequency difference between two beams, is tunable in the AOM DMI [1], frequency content of the measurement signal is reported for two

different beat frequencies that are used in commercially-available phase measuring electronics.

Figures 7(a) and 7(b) show the frequency content for constant velocity target motion of 10,000 mm/min for beat frequencies of 3.64 MHz and 20 MHz, respectively. The Doppler frequency for the selected velocity is 0.53 MHz. Only the intended *ac* interference signal at the Doppler up-shifted frequency for the selected motion direction (+x in Fig. 2) is seen. No extra peaks (i.e., leakage induced interference terms) are observed.

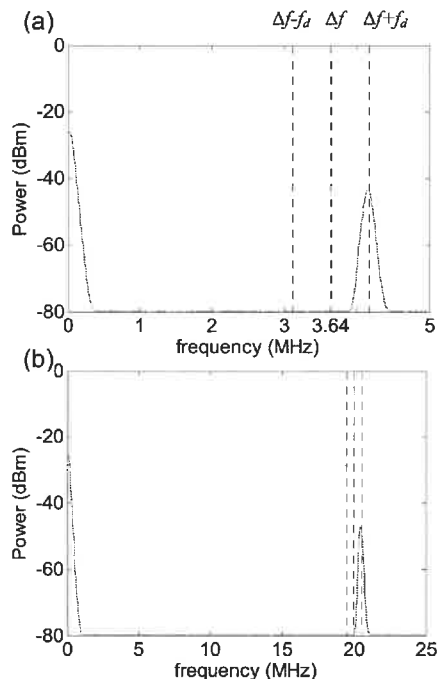


FIGURE 7. Frequency content of the measurement signal during constant velocity motion of 10,000 mm/min ($f_d = 0.53$ MHz) at beat frequencies, Δf , of (a) 3.64 MHz; and (b) 20 MHz. Only the intended *ac* interference and optical power terms are present.

CONCLUSION

This study demonstrated the reduced size acousto-optic modulator-based heterodyne displacement measuring interferometer. A right angle prism with a hole was used to modify the beam paths. The footprint area was reduced by 65% compared to the original setup. The frequency spectrum for the new setup shows no content at periodic error frequencies for two different beat frequencies.

ACKNOWLEDGMENTS

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CONSTANT TEMPERATURE/"AIR-REFRACTIVE-INDEX" CHAMBER FOR PRECISE INTERFEROMETER

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INTRODUCTION

The precision of sub-nanometer order or less in displacement measuring interferometry is required due to the progress of nanotechnology^[1]. Numerous commercial displacement optical interferometers using heterodyne^[2] and homodyne^[3] techniques are widely used in the semiconductor industry. However, environmental fluctuations, such as the thermal expansion of optical devices, change in the refractive index and so on, cause nm-order error of displacement measurement. Therefore, to achieve sub-nanometer precision of displacement interferometers, it is required to minimize the influence of these environmental fluctuations. In the normal air environment, air refractive index fluctuation (Δn_{air}) is a dominant factor causing the measurement uncertainty. This kind of environmental influence can be compensated passively by numerical methods given by Edlen^[4] or Ciddor^[5] using environmental sensors. These methods are quite simple but the measurement speed is relatively low due to the time response of environmental sensors, and resolution is limited to 10^{-8} -order^[6]. It is not satisfied to the requirement of sub-nanometer displacement precision. Recently, Fabry-Perot cavity (FPC) has been utilized with ultralow thermal expansion material (ULTEM) and Pound-Drever-Hall method^[7] for a passive compensation of Δn_{air} in displacement measurement with the uncertainty of 10^{-9} -order^[8]. We would like to make a measuring system with stabilization of Δn_{air} that can be applied to any kind of precision optical interferometer. For this reason, an active compensation of Δn_{air} is preferable. For the active compensation of Δn_{air} , a vacuum environment is effective. However, the vacuum chamber has the problem of high cost. In addition, the experiment may not be possible because of the material characteristics in the vacuum environment.

This paper proposes a construction of a constant temperature/"air-refractive-index" chamber using a FPC in which the stabilities for

temperature and air refractive index are 1 mK and 10^{-10} -order, respectively, in normal air environment. In the chamber, Δn_{air} will be suppressed by the actuation of the pressure (or volume) of the constant-temperature chamber by maintaining a specific dark (or resonance) fringe of the FPC, which is supported by an ULTEM inside the chamber. The Pound-Drever-Hall method is applied to control the chamber volume actuator to compensate Δn_{air} . With the stable temperature and air refractive index inside chamber, a precision interferometric measurement instrument can be operated in the chamber with the influences from environmental effects to the measurement uncertainty is less than 10^{-10} -order.

MEASUREMENT PRINCIPLE

Figure 1 shows the proposed constant temperature/"air-refractive-index" chamber. The air refractive index inside the chamber can be assumed as a function of pressure (p), temperature (T), humidity (h) and CO_2 concentration (CO_2) $n_{air} = n_{air}(p, T, h, \text{CO}_2) \approx n_{air}(p, T)$ with an assumption that the effects of humidity and CO_2 concentration in the experimental room environment are neglected. Therefore, the Δn_{air} is calculated from pressure and temperature changes Δp , ΔT by the equation

$$\Delta n_{air} = \frac{\partial n_{air}}{\partial p} \Delta p + \frac{\partial n_{air}}{\partial T} \Delta T \quad (1).$$

If ΔT and Δp can be controlled to be zero then the Δn_{air} would be approximate zero. In figure 1, the chamber is connected with the constant temperature water bath through the pipes, which are roundly covered the chamber wall. The water flows around the chamber through the pipes circularly to keep the temperature on the chamber wall to be stabilized. The environmental sensors (ES) such as temperature and pressure sensors are also installed to monitor the inside temperature and pressure of the chamber, respectively. A digital signal processor (DSP) with analogue to digital

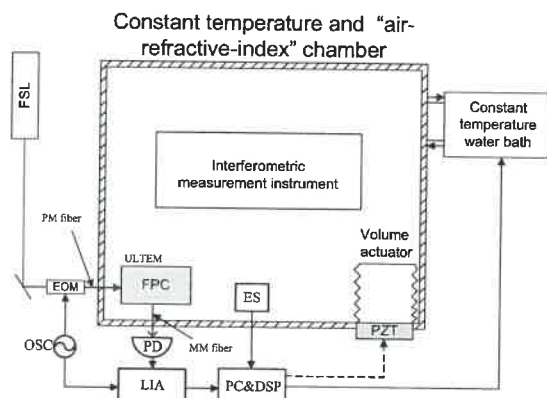


FIGURE 1. Design of the proposed chamber. FSL: frequency stabilized laser, EOM: electro optical modulator, FPC: Fabry-Perot cavity, OSC: oscillator, PD: photo detector, LIA: lock-in amplifier, ULTEM: ultra low thermal expansion material, ES: environmental sensors, PC&DSP: personal computer and digital signal processor.

and digital to analogue converters uses ES signal control the temperature of the water in the constant temperature water-bath device. The inside temperature stability is expected to be 1mK or less^[9], so that, the influence of the temperature to the air refractive index is neglected.

We assumed that the chamber is completely closed (or no leakage of air from inside to outside the chamber). The relationship of pressure and the volume of the chamber can be explained by the ideal gas law as following equations,

$$pV = mRT \quad (2),$$

or

$$\frac{\Delta p}{p} = -\frac{\Delta V}{V} + \frac{\Delta T}{T} \quad (3),$$

where V , m and R are volume, number of moles and air constant, respectively. Δp , ΔV and ΔT are shifts of p , V , T , respectively. Equation (3) shows that when the temperature inside chamber is kept constant, the inside pressure of the chamber is changed following the change of the chamber's volume. Therefore, we can compensate Δn_{air} by adjusting the pressure inside the chamber. A volume actuator, driven by the PZT or other actuator, is set in the chamber (see figure 1) to adjust the inside air pressure.

In order to control properly the volume of the chamber, Δn_{air} needs to be observed precisely. In the chamber, a FPC, supported on an

ULTEM, is installed as a Δn_{air} sensor. The laser beam from a frequency stabilized laser (FSL) is used as the light source for the FPC. The beam is phase-modulated by an electro optical modulator (EOM) and fed into the FPC via a polarization maintaining single mode fiber (PM fiber). Reflection light from FPC is measured by a photodiode (PD) via a multi mode fiber (MM fiber). The PD converts the power of the modulation light to RF signal and a demodulation process is performed by a lock-in amplifier (LIA). At the resonant state of the FPC, the output of the LIA is null^[7], and the following condition is satisfied

$$n_{air}L = \frac{\lambda}{2}N = \frac{c}{2f}N \quad (4),$$

or

$$\frac{\Delta n_{air}}{n_{air}} = -\frac{\Delta f}{f} - \frac{\Delta L}{L} + \frac{\Delta N}{N} \quad (5),$$

where n_{air} , L , λ , f , N , and c are the air-refractive-index, the distance between two cavity mirrors, the vacuum wavelength of the FSL, the frequency of the FSL, the interference fringe order and the speed of light, respectively. Δn_{air} , ΔL , Δf and ΔN are the shifts or fluctuations against n_{air} , L , f and N , respectively. In equation (5), with the use of the FSL and ULTEM, the shifts Δf , ΔL can be neglected. Therefore, Δn_{air} can be derived from the interference fringe number N .

$$\frac{\Delta n_{air}}{n_{air}} = \frac{\Delta N}{N} \quad (6).$$

When the air refractive index inside chamber is changed, the off-resonance of the FPC is occurred. It makes the LIA signal is not a null value. A DSP with the analogue to digital and digital to analogue converters uses the LIA signal to generate a compensation signal for controlling the chamber volume actuator to keep the resonance state of the FPC.

PRELIMINARY EXPERIMENT

To construct and test the constant temperature/"air-refractive-index" chamber, two procedures are required: the first is the precise measurement of Δn_{air} inside chamber. The second is to compensate Δn_{air} by controlling chamber's volume. For the first step of the construction, we have performed the measurement of Δn_{air} using a combination of the FPC, ULTEM and an external cavity laser diode (ECLD) which is used as the light source for the

FPC. In the second step of the construction, we will use the FSL.

The optical schematic of the Δn_{air} measurement is shown in figure 2. In this first step, the signal output of the DSP is not used to control the bellows actuator (as drawn in figure 1) but it is used to control the ECLD frequency (as shown in figure 2). When the distance between two cavity's mirrors is equal to a number of half-wavelength of the ECLD, the resonance of the FPC is occurred. At this state, the LIA signal crosses a null point. Because of Δn_{air} , the wavelength of the ECLD in air is changed. It causes the FPC is out of the resonance (or the output of the LIA is not a null value). The ECLD frequency is controlled by DSP with some analogue/digital interfaces to maintain the resonance of the FPC. Therefore, Δn_{air} can be derived from the ECLD frequency shift.

$$\frac{\Delta n_{air}}{n_{air}} = -\frac{\Delta f}{f} \quad (7),$$

where f , Δf are center frequency of the ECLD and the change of the ECLD frequency, respectively.

The ECLD (New focus, Velocity 6304, wavelength 633 nm, a linewidth of 300 kHz) is phase-modulated by the EOM (New focus, EOM 4002), then fed into the FPC. The FPC is formed from one flat mirror (reflection ratio of 99%) and one curvature mirror (radius: 400 mm, reflection ratio of 99.4%) which are mounted on the ULTEM (Nippon Steel Corp. NEXCERA, $2 \times 10^{-8} \text{ K}^{-1}$) the optical path between two mirrors is approximately of 100 mm. Under the temperature change of less than 50 mK, the optical path deformation between two mirrors is less than 10^{-9} -order. The reflection laser from the FPC is measured by photodetector (Hamamatsu Photonics Co., Avalanche photodiode: C5658), and the photodetector signal was demodulated by the LIA, which includes a passive mixer (R&K, M1XCA double balanced mixer) and a low pass filter (NF Electronic Instrument E-3201, cutoff frequency = 1 kHz). The ECLD frequency shift was obtained by measuring the beat signal between the frequency stabilized laser (FSL, Spectra Physics, 117A He-Ne laser) and the ECLD. A high speed PD (New Focus, high-speed photodetector: 1434) and a frequency counter (FC, Pendulum, CNT-90) are utilized for the beat frequency measurement.

The optics, as shown in figure 2, is set on the anti-vibration optical table with a cover box to reduce the environmental fluctuation. The FPC

is also covered by a smaller shield box which includes heat-isolation material mounted with a copper wall. The temperature, pressure and humidity inside the shield box are approximately of 8[mK], 8[Pa] and 1[%], respectively. A thermometer (Takara Thermistor: DS103), a

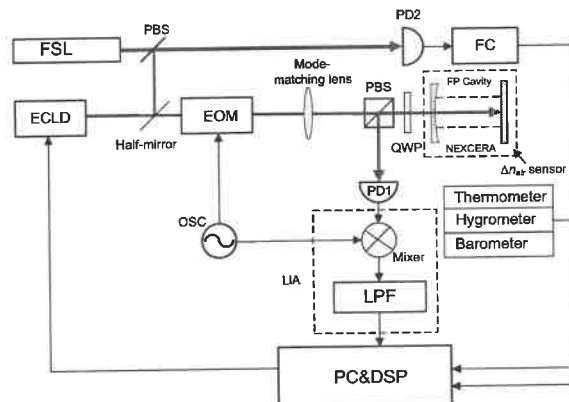


FIGURE 2. Schematic of optical setup for Δn_{air} measurement. FSL: frequency stabilized laser, PBS: polarization beam splitter, OSC: oscillator, FC: frequency counter, LPF: low-pass filter

barometer (Tokyo Suzuki: T-60), and a hygrometer (Delta OHM, HD2101.1) are used to calculate $\Delta n_{air}/n_{air}$ by Ciddor calculation^[6] as a reference to compare with the proposed method. The comparison of the $\Delta n_{air}/n_{air}$ measurement using the FPC and Ciddor methods is shown in figure 3.

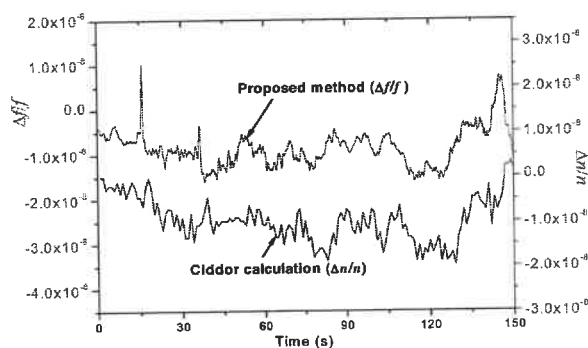


FIGURE 3. Comparison of $\Delta n_{air}/n_{air}$ measurement by the FPC and Ciddor methods under quasi-stable environmental condition

In figure 3, the variation obtained with the proposed method is similar to that obtained by the Ciddor method. A cross-correlation of the two results is shown in figure 4 below. The cross-correlation coefficient is approximately 0.90. It means that the Ciddor and our proposed methods are strongly related. From the result, the measurement uncertainty of the proposed

method is estimated. The uncertainty of $\Delta n_{air}/n_{air}$ measurement depends on three factors: the uncertainty of laser frequency stability, the control quality of the fringe number N and the deformation of the optical path length L . The estimation uncertainty of the method can be represented by

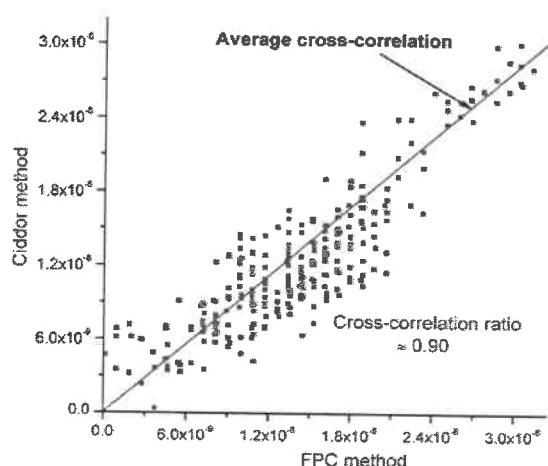


FIGURE 4. Cross correlation plot of the proposed and the Ciddor methods

$$U_{\Delta n_{air}/n_{air}} = \sqrt{\{U_{\Delta f/f}\}^2 + \{U_{\Delta L/L}\}^2 + \{U_{\Delta N/N}\}^2} \quad (5),$$

where $U_{\Delta f/f}$, $U_{\Delta L/L}$ and $U_{\Delta N/N}$ are the uncertainties of $\Delta f/f$, $\Delta L/L$, and $\Delta N/N$, respectively. In the experiment, $U_{\Delta f/f}$ is estimated approximately of 1×10^{-9} -order according to the laser specifications. $U_{\Delta L/L}$ in the experiment is approximately of 3×10^{-10} -order according to the temperature fluctuation during the measurement time. $U_{\Delta N/N}$ in the experiment is approximately of 8.4×10^{-10} -order according to finesse (approximate of 40) and free spectral range (FSR) of the cavity (approximate of 1.5 GHz). The total uncertainties in $\Delta n_{air}/n_{air}$ measurement by the proposed method is summarized in table 1. We can see that if the other laser (such as iodine frequency stabilized He-Ne laser) that has better frequency stabilization, then the $U_{\Delta f/f}$ can be improved to 10^{-10} -order or less. Therefore, we can monitor the $\Delta n_{air}/n_{air}$ with the resolution of 10^{-10} -order.

TABLE 1. Total uncertainties of $\Delta n_{air}/n_{air}$ measurement using FPC method

Uncertainty	$\Delta f/f$	$\Delta L/L$	$\Delta N/N$	Total
Proposed method	1.1×10^{-9}	3.2×10^{-10}	8.4×10^{-10}	1.4×10^{-9}

CONCLUSION

We have proposed a construction of a constant temperature/“air-refractive-index” chamber using optical cavity in which the stabilities for temperature and air refractive index are 1 mK and 10^{-10} -order, respectively. The chamber is suitable for precise interferometric measuring systems. The first step of the construction to measure the air-refractive-index fluctuation was done with the measurement uncertainty of 10^{-9} -order. It is possible to improve the measurement uncertainty to 10^{-10} -order if a better laser source is used instead of the ECLD. From this precise monitor, precise compensation of Δn_{air} will be performed using chamber’s volume actuator.

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ROBUST GATE SYSTEM DESIGN FOR PRECISION SEMI-SOLID CASTING PROCESSES

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ABSTRACT

Semi-solid casting (SSC) of magnesium alloys is increasingly being used to produce high quality components. Using this process, higher strength, thinner wall section and tighter tolerance without porosity are obtained. High-precision fabrication of thin-walled components with large surface area is possible. They are widely used for the IT, auto and consumer electronics industries. However, warpage degrade quality of the part produced in the SCC process. To produce thin-walled Mg alloy parts, the geometry of gating system affecting quality of the product should be thoroughly studied. In this paper, to minimize warpage of the thin-walled sections, Taguchi method is applied to the optimal design of the gate geometry in the thixomolding process. Width, height, length and angle of the gating system are selected for the robust design parameters. Effectiveness of the robust design is verified through the CAE software.

Key Words: Gate system, Mg alloy, Semi-solid casting, Taguchi method, Thixomolding, Warpage

INTRODUCTION

Magnesium has excellent electromagnetic interference (EMI) shielding, good damping, high specific strength characteristics. It is widely used for the IT, auto and consumer electronics industries [1,2]. In general, Mg parts have been produced through the die casting process. Recently there is increasing demand to apply semi-solid casting (SSC) of Mg alloys for manufacturing of high performance components. This process is similar to the injection molding of plastics with very high pressure and is called thixomolding. It is strictly governed by thixotropic property of injected semi-solid slurry. Using this process, higher strength, thinner wall sections and tighter tolerances without porosity are obtained. High specific strength characteristics of magnesium alloys render high-precision

fabrication of thin-walled components with large surface areas. However, warpage of the thin-walled section degrade quality of the part produced in the SCC process. To understand the thixomolding process of AZ91D Mg alloy parts, mechanical properties and dimensional accuracy of the thixomolding process were studied in the previous work [3]. As the microstructure of thixomolding product consists of spheroidal particles compared with dendrites of conventional casting product, and the injection pressure is much greater than that of the die casting process, SSC manufactures high-integral and low-porous products. It produces better mechanical property than what is achievable by the conventional liquid metal die casting process. However, there is difficulty for maintaining coexistence of liquid and solid in the thixomolding process. An inadequate set of process parameters causes high porosity and unequal cooling in the thixomolding. Also, it causes deformation of products [3,4]. In the previous study, optimal parameters of injection velocity, die temperature, barrel temperature, and so on affecting the quality of the finished products were studied when the gate system had been specified by an expert [3].

In addition to the previous study, inadequate geometry of the gating system generates turbulent flow and cavity during the SSC process. Irregular cooling deforms the product as well [5]. For the better product quality the geometry of gate system of the SSC mold should be studied thoroughly. In this paper, Taguchi method is applied for optimal design of the gate geometry in the thixomolding process when the operating parameters are fixed to the previously obtained optimal values of injection speed, barrel temperature and mold temperature [3]. Optimal width, height, length and angle of the gating system are studied for the robust design parameters. Effectiveness of the robust design is confirmed through the CAE sysem, AnyCasting [6].

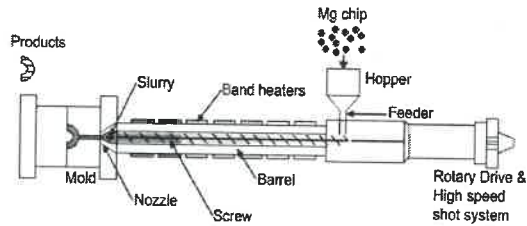


FIGURE 1. Schematic diagram of thixomolding process.

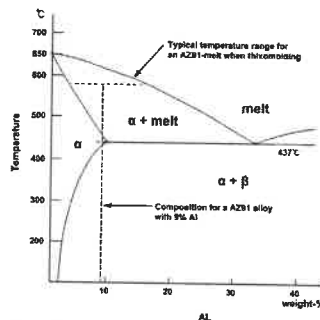


FIGURE 2. Mg-Al phase diagram for the thixomolding process.

THIXOMOLDING PROCESS

Fig. 1 shows a schematic diagram of the thixomolding process. 3~4mm chips are put into the hopper like the injection molding. To achieve the required solid to liquid ratio of the slurry, injection temperature is selected based on the liquidus and solidus values of the Mg-Al phase diagram shown in Fig. 2 [2]. In this paper, AZ91D Mg alloy is selected to fabricate a cell-phone battery cover.

TAGUCHI METHOD

Taguchi method employs signal-to-noise (SN) ratio to quantify experimental variation. In this paper, SN ratio for smaller-the-better (SB) given by Eq.(1) is applied for reducing warpage of the product.

$$SN = -10 \log \left[\frac{1}{n_s} \sum_{i=1}^{n_s} y_i^2 \right] \quad (1)$$

$$M_{Ai} = \frac{1}{n_A} \sum_{j=1}^{n_A} [(SN)_{Ai}]_j \quad (2)$$

In Eq. (1) n_s is the total number of characteristic values in the orthogonal array (OA) table. y_i is the i th characteristic value. Mean statistical analysis is used to obtain optimal combination of design parameters, and to enable robust design of parameters. In Eq. (2) M_{Ai} is the mean SN ratio of the factor A at level i , and n_A is the number of

appearances of the factor in the OA table. $(SN)_{Ai}$ is the SN ratio of factor A at level i , and subscript j indicates the level number.

Analysis of variance (ANOVA) is a collection of statistical models, and their associated procedures, in which the observed variance is partitioned into component due to different explanatory variables. Fundamental technique is a partitioning of the total sum of squares into components related to the effects used in the design experiments [3]. SN ratios are derived from Eqs. (1) and (2). Variances of each factor are applied to analyze contribution of the factor in the thixomolding process.

ROBUST DESIGN OF GATE SYSTEM

AnyCasting CAE system [6] is used to simulate the thixomolding process for making battery covers of a cell-phone. Mg alloy AZ91D with a nominal composition of 8.5% Al, 0.75% Zn, 0.3% Mn, Fe and Ni below 0.001%, and Mg as a balance is used for the SSC process. The casting quality depends upon design parameters

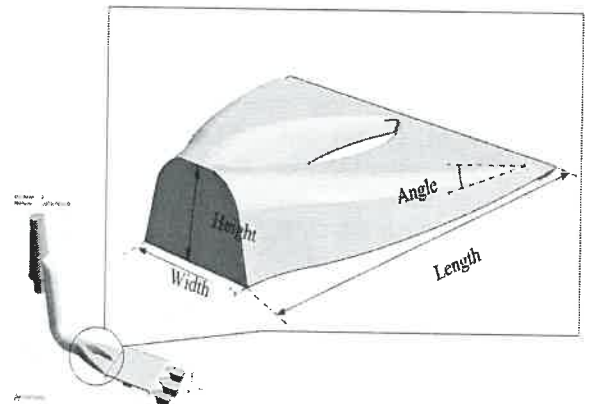


FIGURE 3. Definition of design parameters for robust design.

Table 1. Design parameters and levels (mm).

Level	Width(A)	Height(B)	Length(C)	Angle(°) (D)
0	13.128	8	40	9
1	12.472	7.6	39	8.55
2	11.816	7.2	38	8.1

TABLE 2. Orthogonal array of design parameters.

No.	A	B	C	D
1	0	0	0	0
2	0	1	1	1
3	0	2	2	2
4	1	0	1	2
5	1	1	2	0
6	1	2	0	1
7	2	0	2	1
8	2	1	0	2
9	2	2	1	0

A: Width, B: Height, C: Length, D: Angle

of the gate system. The objective of robust design is to derive optimal gate geometry to improve flatness of the SCC part located between the gates and overflows in the mold assembly. Fig. 3 shows CAD model of the product and the gate system. Design parameters are defined by the width, height, length and angle of the gate entry. Table 1 shows 4 design factors with three levels selected through prior experiments [3]. Table 2 shows OA table $L_9(3^4)$ with respect to design conditions selected in Table 1.

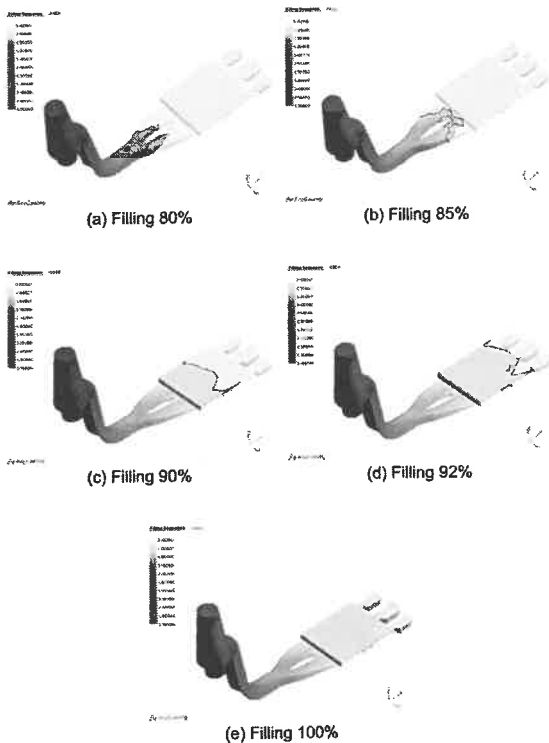


FIGURE 4. Filling procedures of AZ91D in the SCC process.

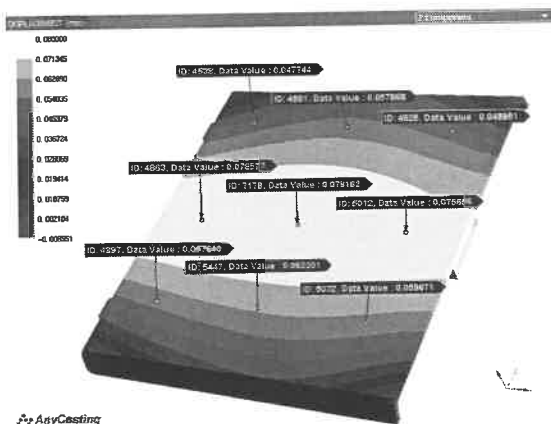


FIGURE 5. Flatness measurement points in the warpage analysis.

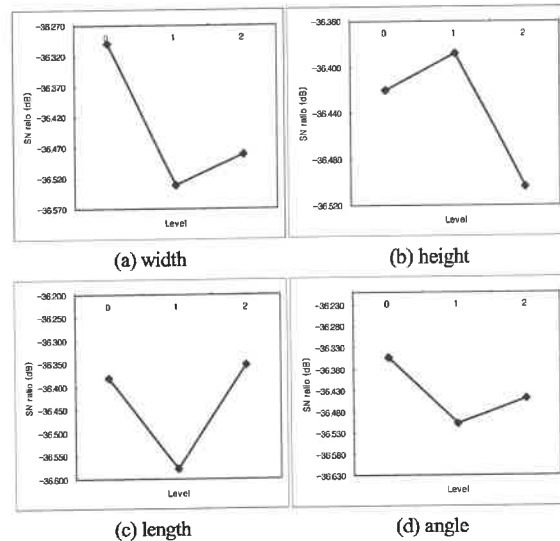


FIGURE 6. Factor response graph of design parameters.

CAE analysis and results

Assessment of flatness is important to fabricate aesthetic and functional IT and digital appliance products. Warpage degrades flatness. This is generated due to residual stresses occurred in cooling process. Thixomolding simulation with best operating parameters obtained in the previous research [3] has been conducted through AnyCasting according to 9 kinds of numerical simulation conditions described in Table 2. Fig. 4 shows filling procedures in the SCC process. Irregular filling is observed during the process. It generates warpage of the part during the cooling cycle. Fig 5 shows 9 measurement points for estimating the flatness of the cell-phone battery cover. It confirms that white region deforms too much comparing with 4 corners. Table 3 shows 9 measurement results, average values of the warpage and SN ratios according to design conditions. The SN ratios are computed by Eq. (1). Fig. 5 shows the mean SN ratio with respect to each design factor, which are computed by Eq. (2). It reveals optimal design parameters to minimize the warpage, which are 13.128mm width, 7.6mm height, 38 mm length and 9° angle.

Table 4 shows ANOVA analysis [3] results. Width of the gate system (factor A) has 37.61%, length (factor B) has 38.18%, and angle (factor D) has 15.19% contribution to the warpage. This confirms that the width and length of the gate system significantly affect the flatness. They should be carefully selected to reduce warpage error in the SCC process.

Table 3. SN ratios of the flatness measurement in the robust design process (unit: μm).

No.	Measurement points									Average Value	SN ratio
	1	2	3	4	5	6	7	8	9		
1	47.744	57.868	48.951	78.577	79.162	75.656	57.640	63.201	59.671	63.163	-36.147
2	50.923	55.802	51.126	82.043	83.218	76.130	61.906	62.604	65.809	65.507	-36.462
3	49.971	57.850	51.173	80.183	81.367	75.259	61.050	62.794	58.956	64.289	-36.292
4	50.908	60.786	51.111	85.003	83.174	80.092	63.863	66.546	61.765	67.028	-36.667
5	50.881	55.286	52.061	81.650	83.645	74.801	60.818	62.258	57.715	63.346	-36.314
6	50.980	60.865	51.184	85.186	83.371	78.263	63.048	65.790	60.954	66.627	-36.614
7	50.842	57.709	51.044	83.862	82.023	77.959	61.730	63.374	59.629	63.352	-36.450
8	49.958	58.841	50.163	83.134	81.313	76.214	60.994	63.718	59.899	64.915	-36.388
9	50.944	60.827	51.148	85.096	83.276	78.179	62.959	65.676	60.863	66.552	-36.605

In order to confirm the robust design result, CAE simulation has been conducted by using the optimally selected design parameters. The SN ratio and average warpage are obtained as -36.027dB and $62.281\mu\text{m}$ through the robust design, respectively. It has better flatness than other 9 gate cases as shown in Table 3. The optimal gate system reduces its volume by 10% as well. This saves production cost too.

TABLE 4. ANOVA results of design parameters.

Factor	SS	DOF	V	F0	P(%)
A	0.09	2	0.044	4.17	37.61
B	0.09	2	0.045	4.23	38.18
C	0.04	2	0.018	1.69	15.19
Error	0.02	2	0.011		9.02
Total	0.24	8	0.118		100

CONCLUSIONS

Robust design of the gate system in the SSC process of AZ91D Mg alloy has been conducted to fabricate thin-walled battery cover of a cell-phone. To improve flatness and reduce material cost, $L_9(3^4)$ OA table and Taguchi method have been applied for selection of optimal design parameters of the gate system. AnyCasing CAE system has been used to derive the following conclusions:

1. Flatness has been improved through the robust design of the gate system.
2. ANOVA results show that flatness is affected significantly by the width (37.61%) and length (38.18%) of the gate system.
3. Minimal flatness is obtained when the gate system is designed with 13.128mm width, 7.6mm height, 38 mm length and 9° angle.
4. Volume of the gate system has been reduced 10% comparing with the conventional design. This saves production cost.

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PRECISION EMBOSSING OF MOLDS AND DIES BY AUTOMATIC BOUNDARY TRACKING OF COMPLEX IMAGES

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ABSTRACT

A precise reverse engineering method generating 2.5D embossed models from images is proposed through the 2.5D reconstruction methods. To generate 2.5D TechArt models from the artistic images, edges are extracted through the image preprocessing and automatic boundary following techniques. Accurate NURBS interpolation of boundary pixels generates TechArt CAD models well. Performance of the developed system has been confirmed through the quick turnaround 2.5D engraving simulation linked with the commercial CAD/CAM system.

Key Words: 2.5D embossing, Aesthetic images, Dies and molds, Image processing, NURBS, TechArt

INTRODUCTION

Recently, aesthetic and artistic design is important for upgrading quality of products and manufacturing high value-added goods. This generates TechArt technology in the IT, digital appliance, and auto industries. TechArt means a design technology to transform engineering items to products with aesthetic design features. Using the artistic design patterns, design quality of commercial products is significantly upgraded. Valuation of IT, digital appliance and auto products is improved as well [1].

Since most image patterns are complex, screen printing of 2D application has been applied for implementation of complex images so far. On the other hand, the precision reverse engineering of 2.5D and 3D replicas of the complex images has been not only time-consuming processes for CAD developers, but also requires much lead-time for the application of product design process now. Several kinds of contact and non-contact 3D digitizers linked with specific surface modelers have been applied to construct 3D CAD data from a real model. However, they are expensive, inaccurate and complex for application of the product design as well.

In this paper, an accurate and rapid reverse engineering tool applicable to construction of embossed CAD data from any kinds of artistic pictures or images is studied. Automatic linking of edge pixels generates boundary curves of the images. To smooth and modify the identified boundary curves, NURBS interpolation is conducted. In addition, to maximize the interfacing flexibility with commercial CAD/CAM systems, IGES translation is applied for the CAD/CAM interface. Embossed 2.5D artistic CAD models for aesthetic fabrication of molds and dies are then generated from 2D complex images through the 2.5D reconstruction method. Performance of the developed system is verified through the CAD/CAM system.

IMAGE PROCESSING

Images are contaminated by noise. It deteriorates quality of reconstructed 2.5D CAD models. To reduce the noise, color images are transformed to gray images first. To remove impulse and salt-and-paper noises without blurring of the image, an adaptive median filter [2] is applied to the image. Then binary image is obtained through multiple threshold according to designer's intention. In the application of the adaptive median filter, selection of the proper window size is important to enhance clearness of the image. Fig. 1 shows the image processed binary image.

AUTOMATIC BOUNDARY FOLLOWER

To get 2.5D solid CAD models from an image, edge information should be extracted from the image first. In this paper, the automatic boundary follower (ABF) is devised from the simple boundary following (SBF) algorithm [3]. Fig. 2 describes it tracing a boundary curve from the start point *P* in the clockwise (CW) direction. If intensity 1 pixel (gray in Fig. 2) is detected when the ABF is scanning a binary image from the top left pixel to the right direction, a follower *S* is triggered at pixel *P*. It moves to upward, and then turns to the right direction for the CW trace when

the upper pixel has intensity 0. In addition, if the follower meets a pixel with intensity 1, it turns to the left direction. Then the follower E finally comes to the start point P and ends. By connecting the intensity 1 pixels along the tracking direction, a boundary curve of an object O is obtained. This is called a real loop. However, the follower does not make a boundary loop at P because of 0 intensity pixel along the tracking path. This is called a point loop.

Fig. 3 shows a recognition process of a twisted boundary curve connecting objects O_1 and O_2 . Pixels $P_1, P_2, P_3, P_4, P_5, P_6, P_7$ and P_{10} make point loops. Pixels P_8, P_9 and P construct real loops. The real loop P_8 recognizes pattern O_2 , and the pattern O_1 and twisted boundary curve are recognized through the pixels P_9 and P .



FIGURE 1. Result of the adaptive median filtering

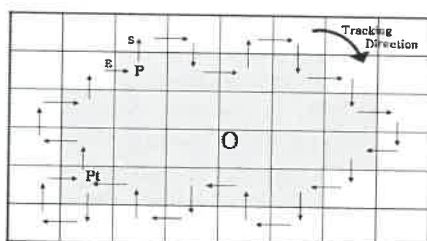


FIGURE 2. Recognition of the boundary curve through the SBF algorithm in the CW tracking.

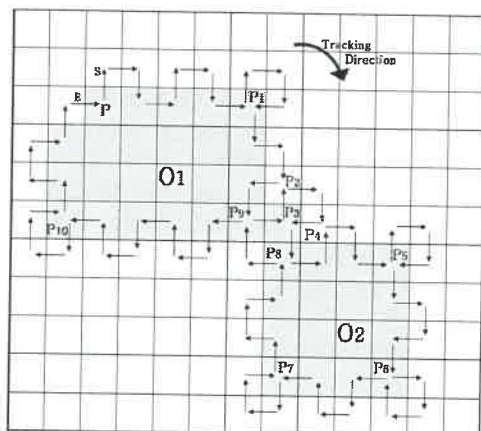


FIGURE 3. Recognition of the twisted boundary curve through the SBF algorithm in the CW tracking.

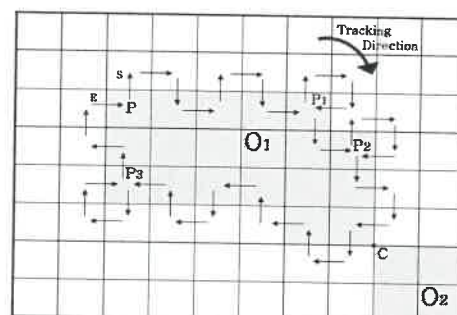


FIGURE 4. Misrecognition of the boundary curve through the SBF algorithm in the CW tracking.

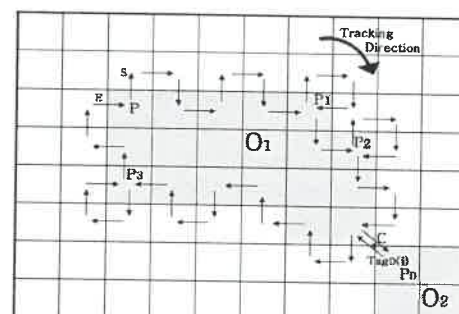


FIGURE 5. Recognition of the boundary curve through the TagD(i) vector in the CW

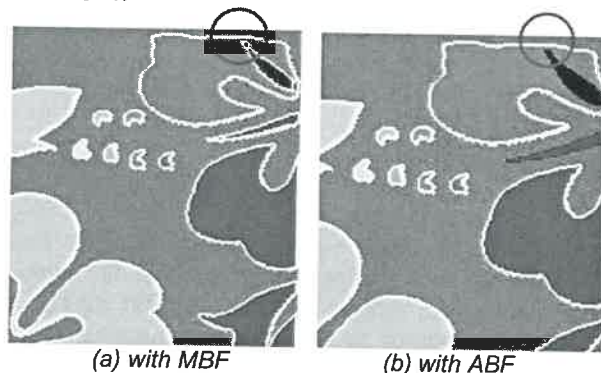


FIGURE 6. Healing of small objects through the ABF algorithm.



FIGURE 7. Closed boundary curves of an aesthetic image through the ABF algorithm.

However, the SBF algorithm does not recognize a twisted curve when two objects O_1 and O_2 are connected at C like Fig. 4. To resolve this misrecognition problem, $TagD(i)$ vector is devised

to monitor object O_2 when the follower turns the trace direction from the right to the left at the bottom right corner of an object. $TagD(i)$ is set to 1 when the intensity of pixel P_D is 1. Then the follower moves to P_D pixel and traces object O_2 in CW tracking direction. $TagD(i)$ is reset when the tracking ends at P_D and the follower returns to O_1 and ends trace at P . This modified boundary following (MBF) algorithm recognizes a twisted curve without misrecognition.

Using the devised MBF algorithm, real loops with closed curves and/or closed twist curves are extracted from a 2D image. However, CAD systems do not construct solid models from the twist curves. To resolve this problem, the automatic boundary follower (ABF) divides the twisted curve into two closed curves through a healing process at pixels P_8 , P_9 and P_D of real loops. During the manual healing process, designers are able to modify the small patterns as well. Fig. 6 (a) shows the recognized boundary curves through the MBF algorithm and the coloring operation. Fig. 6(b) shows the ABF result after healing the small object in the circular mark. Fig. 7 shows extracted aesthetic patterns of Fig. 1 through the ABF and the coloring operation. It is confirmed that the devised AMF algorithm works well to extract closed boundary curves of aesthetic images.

GENERATION OF NURBS CURVES

In order to make 2.5D CAD models applicable to fabricate dies and molds from pixels of the boundary curves, NURBS curve fitting of them is required. NURBS curve with parameter u and order k is given by [4-6]

$$P(u) = \frac{\sum_{i=0}^n w_i P_i N_{i,k}(u)}{\sum_{i=0}^n w_i N_{i,k}(u)} \quad (1)$$

where P_i is the i th control points, w_i is the weight of P_i , and $N_{i,k}(u)$ are blending functions given by

$$N_{i,k}(u) = \begin{cases} 1 & t_i \leq u < t_{i+1} \\ 0 & \text{otherwise} \end{cases} \quad (2)$$

$$N_{i,k}(u) = \frac{u - t_i}{t_{i+k-1} - t_i} N_{i,k-1}(u) + \frac{t_{i+k} - u}{t_{i+k} - t_{i+1}} N_{i+1,k-1}(u)$$

where t_i is the i th knot value. To secure continuity of the NURBS curve, order k is selected 3 in this paper.

Interpolation of NURBS curves

To derive NURBS curve model interpolating $(m+1)$ pixels, Q_i , of the boundary curve, information of the weight w_i , knot value t_i , parametric value of the i th pixel, u_i^p , and control point value P_i should be provided as Eq. (3). In this case w_i is specified by the designer. t_i , u_i^p and P_i should be carefully selected for the better NURBS fitting.

$$Q_i = C(u_i^p) = \sum_{j=0}^m P_j R_{j,k}(u_i^p) \quad (3)$$

where rational function is given by

$$R_{j,k}(u) = \frac{w_j N_{j,k}(u)}{\sum_{l=0}^m w_l N_{l,k}(u)} \quad (4)$$

Applying Eq. (3) to $(m+1)$ pixels, following matrix equation is derived:

$$[R][P] = [Q] \quad (5)$$

Then, control points for the NURBS interpolation are given by

$$[P] = [R]^{-1}[Q] \quad (6)$$

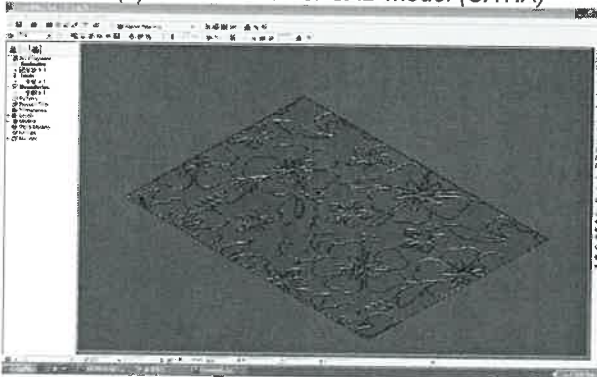
To derive R matrix in Eq. (6), t_i and u_i^p according to Q_i should be selected accurately. In this paper, u_i^p is determined through the chord length method [4-6]. Knot values t_i is computed from the average knot method [4,-6]. Using Eqs. (2), (4), the chord length and average knot methods, and Eq. (6), control points matrix $[P]$ is obtained. Then the NURBS curve of the boundary curve is obtained through Eq. (1).

QUICK TURNAROUND SIMULATION

To verify performance of the developed embossing system, reconstruction of CAD models,



(a) Visualization of CAD model (CATIA)



(b) Machining Simulation (PowerMill)

Figure 8. Machining simulation on the developed 2.5D quick turnaround embossing system.

tool-path generation and machining simulation have been conducted as follows. Applying the devised ABF algorithm and the NURBS interpolation method to the image of Fig. 1, a CAD model has been obtained. To visualize and manipulate the CAD model on commercial CAD/CAM systems, it is converted to IGES files [7]. Fig. 8(a) shows 2.5D solid CAD model with 0.2mm embossment on CATIA. Fig. 8(b) shows the endmill simulation result on PowerMill. As the CAD model is generated and imported to the commercial CAD/CAM system conveniently and accurately, it is confirmed that the TechArt design and application to industrial products is conducted fast and easy through the developed system.

CONCLUSIONS

A precise 2.5D quick turnaround embossing system to apply for the TechArt design and production of quality products has been devised through image processing, automatic boundary following and NURBS interpolation techniques. Following conclusions have been obtained:

1. Adaptive median filter, multiple threshold and binary conversion of images have generated clear images for 2.5D reconstruction of CAD models.

2. Misrecognition of the boundary curve has been resolved by defining the point and real loops, and the modified boundary following (MBF) algorithm.
3. The automatic boundary follower (ABF) divides twisted curves into two closed curves through the healing process, which enables generation of NURBS curves applicable on the commercial CAD/CAM system.
4. Using the chord length and average knot value methods, interpolation of a NURBS curve has been conducted well. IGES interface enables the developed system to be executable on the commercial CAD/CAM system.
5. Quality and performance of the 2.5D quick turnaround embossing system from aesthetic images are verified on the commercial CAD/CAM system.

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CHALLENGES IN ASSEMBLING A MAGNETIC SHAPE MEMORY (MSM) MICROACTUATOR

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ABSTRACT

The development of Magnetic Shape Memory (MSM) materials open up new opportunities in linear micro motion. For enabling a switching motion in a system, a pair of bulk MSM stripes was arranged to operate against each other in a "push-push" mode. For the system to function, the thin-film stators have to be capable of creating a magnetic field strong enough to elongate the MSM material. To achieve the desired field, the air gaps between the stators and the active MSM bars have to be kept to a minimum. For such a system, it is difficult to rigidly stacking the stator, the MSM stripe, and the top stator. This paper discusses the assembly concept: mounting the two MSM stripes inline (with one elongated and one compressed) and spring loading the four stators against them.

INTRODUCTION

Integrating a Magnetic Shape Memory (MSM) material in Micro Electro-mechanical Systems (MEMS) shows a high potential for developing small discrete hybrid microactuators with a large travel range [1]. The traveller of such a system consists of bulk MSM stripes that are excited by a magnetic field generated by the thin-film electromagnetic stators. The main advantage of an MSM material compared to other smart materials (piezoceramics, thermal shape memory, magnetostrictive alloys) is its ability to generate up to 10% strain when exposed to an external magnetic field. This strain is caused by the reorientation of twin variants in the magnetic domain structure. However, such a MSM material may also be set back to its original state mechanically, if a stress greater than the material's blocking stress is applied. To make use of the MSM effect for actuation purposes, a mechanical load may be applied to achieve a

reversible movement [2]. A practical approach is to mount two MSM stripes in a row and elongate one by applying a magnetizing field while the other one is compressed [3]. The MSM stripes are single crystalline NiMnGa with the dimensions of 5 mm x 10 mm x 0.1 mm (Fig. 1).

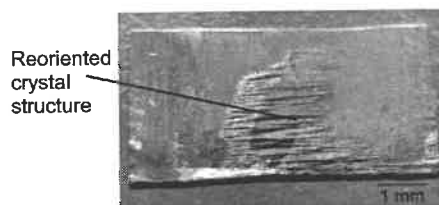


FIGURE 1. Photograph of a bulk MSM stripe

The electromagnetic stators are fabricated in thin-film technology on Al_2O_3 wafers. They feature U-shaped soft magnetic cores and double layer spiral coils wound around each pole. Figure 2 depicts a diced stator chip, its dimensions are 6.3 mm x 0.5 mm x 0.52 mm.

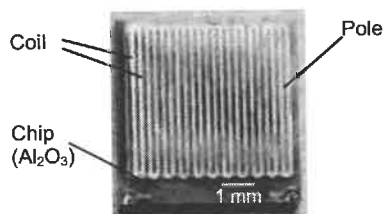


FIGURE 2. Fabricated thin-film microactuator

For allowing a switching motion to take place, a pair of bulk MSM stripes is arranged in line to operate against each other in a "push-push" mode [4]. For causing an elongation, the stripes are exposed to a magnetic field alternately by exciting a pair of electromagnetic microstators located on top and on the bottom of one MSM stripe. Collapsing a stripe into its original state is accomplished by elongating the other MSM

stripe in the line, and vice versa. The appropriate magnetic field for the excitation of the MSM stripe can only be created if the accumulative air gap between the stators and the MSM stripe does not exceed 5 μm . Such an actuator needs an excitation coil system which is able to generate a magnetic field strength of 50 kA/m in the MSM material. This is the field strength necessary for causing a crystalline reorientation of the MSM alloy which causes the elongation of the material [5].

ASSEMBLY APPROACH

Since minimal air gaps are required between the stators and the MSM stripes, a mechanical stacking using spacers during the assembly process is not an option. Such an approach would require creating two in-line stacks, each consisting of a bottom stator, the MSM stripe, and a top stator and to adjust them to each other. What most of all prohibits to taking this approach is the fact that the MSM stripes is warping during their elongation. Thus, another assembly concept was chosen: mounting the two MSM stripes in-line (with one elongated and one compressed) and spring loading the four stators against them (Fig. 3). This way, not only the air gap is minimized, but also any out-of-plane motion of an MSM stripe during actuation can be compensated.

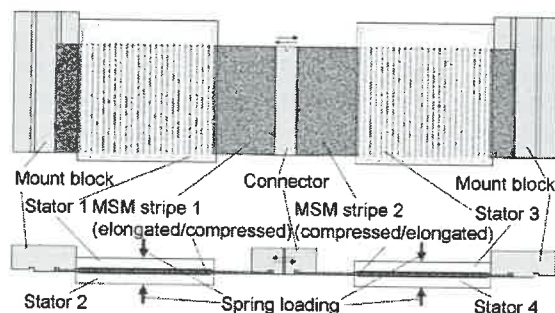


FIGURE 3. Schematics of the assembly concept

To evaluate the assembly concept, a first prototype using spring-loaded push rods was built [4]. While it was functional, the space it took up was prohibitive. Therefore, it was left in favour of an alternative approach using leaf springs. Useful springs were found in flexures used for Hard Disc Drive (HDD) recording heads (Fig. 4).

As first step of the assembly process, the MSM stripes were mounted in a row with a connector between them. For joining the MSM stripes, mount blocks and a connector block were used

(Fig. 5). All blocks were fabricated out of Altic ($\text{Al}_2\text{O}_3\text{-TiC}$) ceramic by dicing.

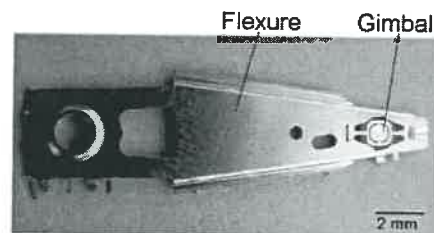


FIGURE 4. HDD flexure

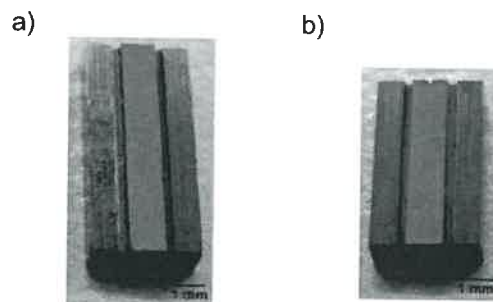


FIGURE 5. System components: a) mount block and b) connector

To achieve a co-planarity and a well defined in-line adjustment of all assembly elements (MSM stripes and stators), a highly precise profiled Altic block serving as a supporting fixture was designed and fabricated (Fig. 6).

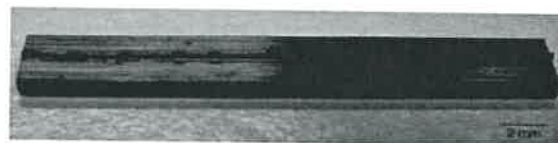


FIGURE 6. Altic supporting block

A further challenge in the assembly was doing a clamping process while the spring force was already applied. To achieve a minimal air gap between the stator's active surface and the MSM stripe, the direction of the spring force application must always be perpendicular to the MSM deformation (elongation or compression). Using HDD flexures for the application of a spring force has many advantages. Besides the much smaller size compared to the push rod version, it also features a gimbal on the side where the force impacts (s. Fig. 4). This assures a self-adjustment of the stators to the MSM surfaces during the actuation even during a warping of the MSM stripes.

ASSEMBLY PROCESS

The first step in the assembly sequence for a prototype hybrid microactuator system was to elongate one MSM stripe using an external magnetic field while compressing the other stripe mechanically. Afterwards, the two MSM stripes (one elongated and the second compressed) were adjusted and bonded in a row, with a connector between them and mounting blocks on both ends. The bonding was executed after aligning the traveller parts and pushing the against the reference plate (Fig. 7). This assembly results in a rigid traveller system with a connector featuring as a moving element (Fig. 8).

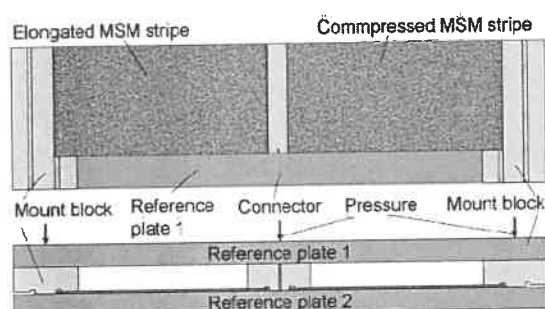


FIGURE 7. Traveler assembly process



FIGURE 8. Row assembly of two MSM stripes with a connector in between

In order to define the height and in-line adjustment of the elements (traveller row assembly and stators), an Altic supporting block was used. Figure 9 shows a completed subassembly. It consists of four HDD flexures (arranged in pairs) and an Altic supporting block between them.

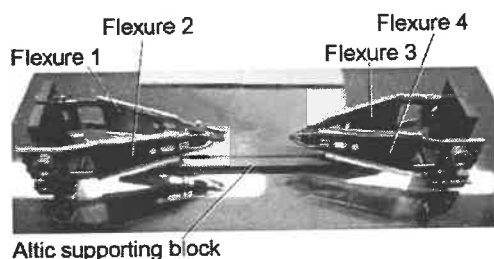


FIGURE 9. Fixture with HDD flexures

To insure a plane contact between flexure gimbal and a stator back side, an Altic block with a footprint of 2.5 mm x 2 mm is bonded on the gimbal of the flexure (Fig. 10a). Afterwards, the traveller row assembly was mounted in the flexure subassembly (Fig. 10b). Next, the stators were mounted on the supported block between the flexures and MSM stripes (Fig. 10c).

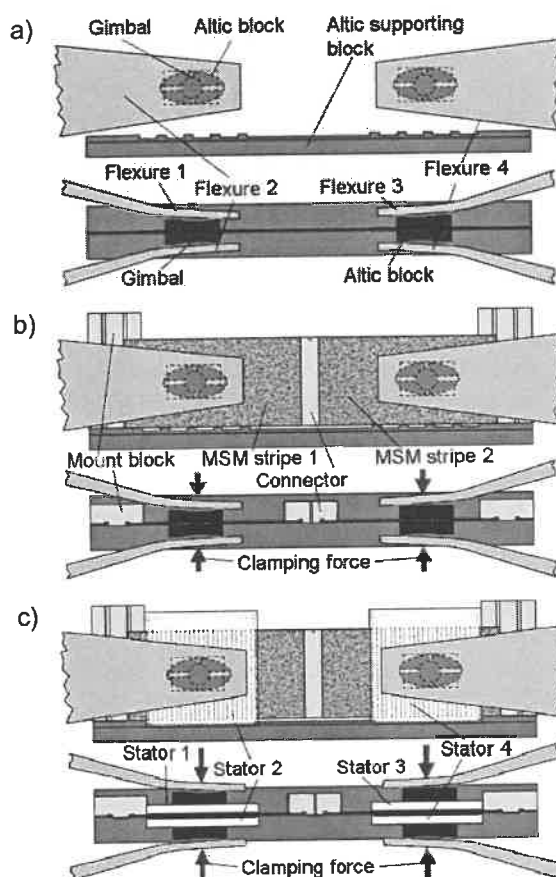


FIGURE 10. Assembly sequences: a) initial state, b) mounting of traveler and c) mounting of stators

The profiles of the Altic supporting block define the position of the assembled elements. The supporting block determines the position of mounting blocks and the MSM stripes (traveler row). Using the profiles of the side blocks, the position of the stators was adjusted appropriately.

The initial load of the flexure determines the clamping force on the stators. On the other side, plane Altic block on the gimbal ensures a perpendicular clamping force regardless of the MSM stripe warping. This way, a small air gap was accomplished without impeding a stripe

elongation. Figure 11 shows an assembly prototype of such a system. The footprint of the MSM system assembly (stators and MSM stripes) is 25 mm x 6.5 mm.

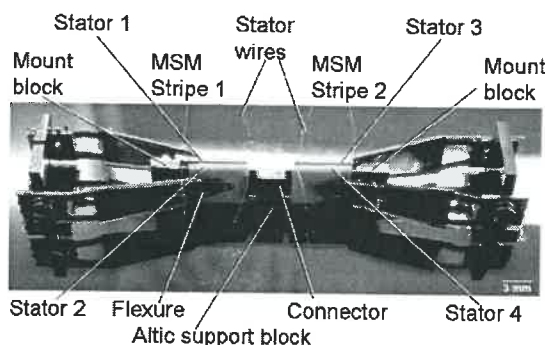


FIGURE 11. Prototype with HDD flexures

The only disadvantage of the flexure system compared to the spring-loaded push rod system is its smaller preload force. This holding force must be predefined by choosing the flexure type or pre-loading it by bending.

CONCLUSIONS

The tight air gap requirements of a MSM "push-push" microactuator and unique expansion characteristics of the MSM material could be accounted for by spring loading the system's thin-film stators to the MSM stripes. For creating the preload force, flexures typically used in HDD heads proved to be an appropriate match. By developing an assembly process, the MSM actuator could be built up.

ACKNOWLEDGEMENTS

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DIAMOND TOOL CENTERING FOR CYLINDRICAL TURNING

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INTRODUCTION

Diamond tool centering for turning using the ogive error [1] method has been extensively used for some time. This method of tool centering is, however, limited to cases where a spherical test part can be machined to include the center of rotation of the part spindle on the part. In the case of cylindrical turning, this is oftentimes not possible or would lead to significant error due to the fact that a completely different portion of the tool would be used for centering than for machining.

High-accuracy tool centering (centering accuracies of the order of $1\text{ }\mu\text{m}$) for cylindrical diamond turning is often necessary for turning cylindrical lenses or mechanical components that require very close-tolerances. One demanding example of a close-fit tolerance is for a mass spectrometer air amplifier system being developed at North Carolina State University. Several components must fit together with cylindrical clearances of $2\text{ }\mu\text{m}$ or less. This means that both the inner and outer diameter tolerances for mating parts is $1\text{ }\mu\text{m}$.

PROCEDURE

The centering method for cylindrical turning involves turning a sphere onto the outer diameter of a cylinder as shown in Figure 1. The tool interpolates a radius in the X/Z plane of the diamond turning machine (DTM) while the rotation of the part around the spindle axis forms an orthogonal radius.

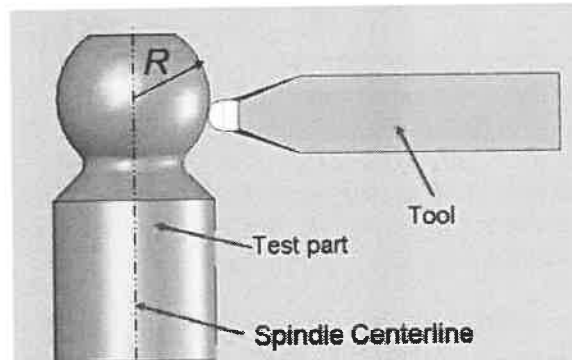


FIGURE 1. The nominally spherical test part for centering a tool nominally normal to the spindle rotational axis.

For an ideal part, these two radii would be equal to radius R in Figure 1. If these two radii are different, however, the result is a toroid with a radius of curvature equal to the toolpath radius minus the tool radius (otherwise known as the interpolated radius) defined as R_i , and a radius of revolution defined as R , as shown in Figure 2. The centers of these two radii are separated by a distance of Δx , the tool centering error.

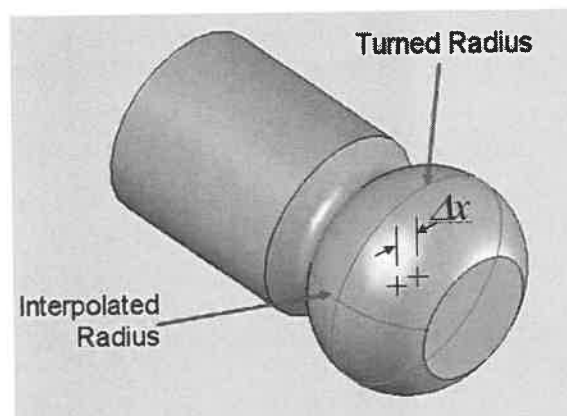


FIGURE 2. The turning of a non-ideal part during the centering operation results in the machining of a toroid with the centers its two constituent radii displaced by a distance of Δx .

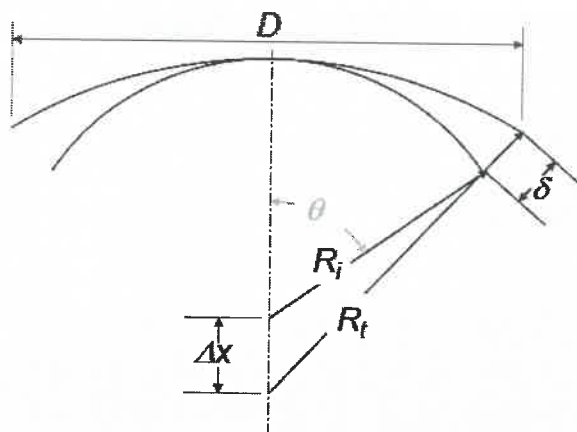


FIGURE 3. A two-dimensional schematic of the parameters involved in determining the tool centering error Δx .

To determine the centering error Δx , the difference, δ , between these two radii can be measured over a defined aperture D as shown in Figure 3. It turns out that this maximum deviation between the two radii as measured in a laser interferometer is equivalent to the magnitude of the astigmatic zernike terms (4th and 5th order). Using this astigmatic magnitude term is an advantage over simply measuring peak-to-valley error in that it is resistant to local variations and other errors in the part.

To derive the tool centering error from the known parameters of R_i , D , and the measured quantity δ , the starting point is defining a triangle as shown in Figure 3 formed by the sides R_i , R_t and Δx . Using the law of cosines and the angle θ ,

$$R_i^2 + 2R_i\Delta x \sin(\theta) + \Delta x^2 = (R_t - \delta)^2. \quad (1)$$

Since is the aperture angle, θ ,

$$\cos(\theta) \approx \frac{D}{R_i} \quad (2)$$

When the difference between R_t and R_i is small. Also,

$$R_t = R_i + \Delta x \quad (3)$$

Since R_t and θ are unknown in Equation 1, by substitution they can be eliminated, the resulting formula then being solved for Δx .

$$\Delta x = \frac{-2\partial R_i + \partial^2}{(-2R_i + GR_i + G\partial)} \quad (4)$$

where

$$G = \sqrt{\frac{D^2 - 4R_i^2}{R_i^2}}. \quad (5)$$

The algebraic solution was determined and checked using Maple™ 12.0.

MEASUREMENT

The aperture D to be measured is shown in Figure 4. The measurement is made over this known aperture to determine the astigmatic magnitude or radial sag difference. The sag difference, δ , can then be directly related to the tool centering error by Eq. 4. Sign is determined from the interferogram. If the surface profile shows the rotation radius R_i , that is normal to the part axis of rotation, to be high at the edges, the tool zero is short of center. If the interpolated radius direction R_t , that is parallel to the part axis of rotation, is shown to be high at the edges, the tool zero is past the spindle centerline.

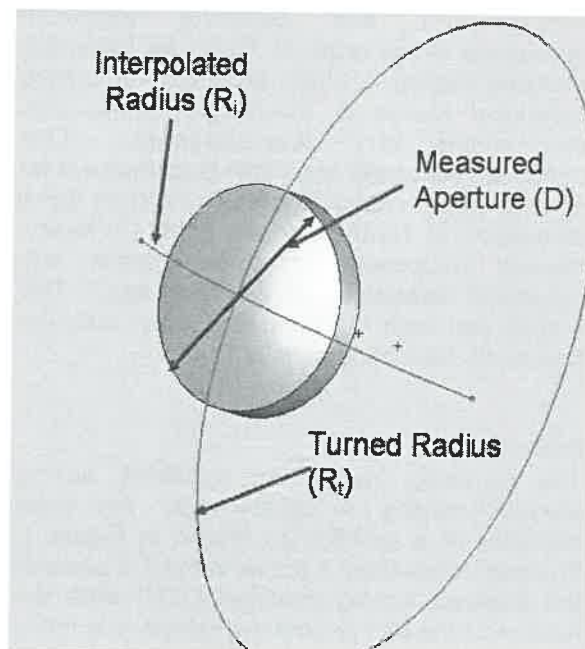


FIGURE 4. Illustration of the aperture D to be measured in the interferometer.

Measurement results of one such test part in copper are shown in Figures 5 and 6 before and after centering correction. The instrument used was a Zygo GPI XP interferometer with MetroPro analysis software. The interpolation radius was 6.3 mm.

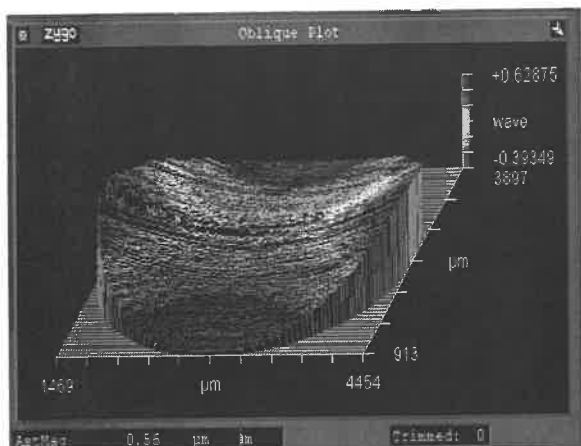


FIGURE 5. Plot of astigmatic error (magnitude: $0.56\ \mu\text{m}$) in a part machined with a tool centering of $35\ \mu\text{m}$.

The measured astigmatic magnitude was $0.56\ \mu\text{m}$, yielding a centering error calculated using Equation 4 of $35\ \mu\text{m}$ on the first sample shown in Figure 5. The Figure also shows the radius in the rotational direction identified by cutting tool marks to be larger than the radius in the interpolation direction. This means that the tool is short of the center of rotation and a corresponding shift of the X-axis coordinate of the DTM must be made. The result of this correction is shown in Figure 6. The residual error magnitude is only 0.02 , yielding a centering error of only $0.3\ \mu\text{m}$. More improvement than this level is unlikely given that the residual bears little of the irregular saddle shape associated with the astigmatic error. Other machining errors dominate in this case.

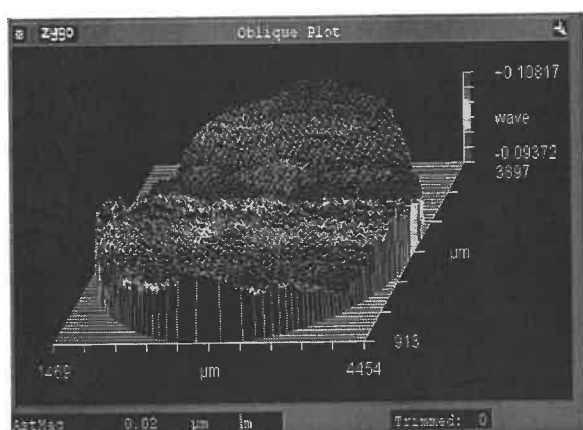


FIGURE 6. Plot of residual astigmatic error (magnitude: $0.02\ \mu\text{m}$) after removal of $35\ \mu\text{m}$ centering error determined from the measurement shown in Figure 5. The remaining centering error is $0.3\ \mu\text{m}$.

CONCLUSIONS

Machining tests have resulted in confirmation of using the described method in practical use, though an independent means of measuring absolute diameter have not yet been obtained. Use in practice for close-tolerance mechanical parts have, nevertheless, shown that accuracies of $1\ \mu\text{m}$ have been obtained for parts of nominally $25\ \text{mm}$ in diameter where interference or loose fit would have resulted from significant error in mating cylindrical parts.

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VALIDATION OF CANTILEVER SHAPE ANALYSIS FOR FORCE MEASUREMENT

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INTRODUCTION

This paper reports the results of calculating an applied force using knowledge of an atomic force microscope (AFM) cantilever's bending shape. The traditional method of measuring force using a cantilever is to measure its deflection, y , at a single point and to relate it to the normal direction force, F , using Hooke's Law, $F = ky$, where k is the calibrated stiffness at that point [1]. Thus, a major component of cantilever force measurement research has focused on this stiffness calibration [2-5]. However, in this scenario, the deflection is affected by lateral forces that arise, for example, from friction at the tip. Using Euler-Bernoulli beam theory to describe a horizontal fixed-free beam with a finite tip height, an improved force relation is:

$$F + \mu \frac{3t}{2L} (F + F_a) = ky(L), \quad (1)$$

where μ is the coefficient of friction, t is the tip height, L is the cantilever length, F_a is the adhesive force between the tip and the sample, and $y(L)$ is the deflection at the free end. Given this model, it becomes impossible to identify the force from a single measurement since the number of unknowns (F_a , F , μ at minimum) exceeds the number of equations.

Now consider a multipoint deflection measurement for the cantilever under load. If the same force equation is applied, but now for the entire length of the beam, the force relation is:

$$y(x) = \frac{-F}{6EI} x^3 + \frac{FL + \mu Ft + \mu F_a t}{2EI} x^2 + 0x + 0, \quad (2)$$

$$= c_3 x^3 + c_2 x^2 + c_1 x + c_0$$

where $y(x)$ is the deflection at any point, x , between the fixed end at $x = 0$ and free end at $x = L$, E is Young's modulus, I is the second

moment of inertia, and the c_1 and c_0 coefficients are nominally zero for a polynomial representation. A least-squares cubic fit to the multipoint measurement would yield estimations of c_3 and c_2 . Then, the applied force, F , could be calculated unambiguously as:

$$F = -6EIc_3. \quad (3)$$

Note that it is not necessary to know the load application location or the beam stiffness at that location in this formulation. All that is required is a physical description of the beam cross section and the material properties. Finally, the shear force, V , in the Euler-Bernoulli beam equation for a uniform cross-section is defined as:

$$\frac{d^3 y}{dx^3} = \frac{-V}{EI}. \quad (4)$$

It follows that $V = F$ for any applied boundary condition. By extension, the shear force in any beam of any shape can be determined by a multipoint measurement over a section of the beam so long as that section follows the assumptions that led to Eq. 4.

EXPERIMENTAL DESCRIPTION

A platform was developed for obtaining multipoint measurements of cantilevers under various loading conditions. The primary component is a scanning white light interferometer, or SWLI (Zygo NewView 7200). This instrument collects topographical information via analysis of the interference pattern between the sample and reference surfaces. Since white light has a short coherence length, there is a large variation in interference intensity over a small change in optical path length. This can be related to the height of points on the sample surface. Other components include stages with translational and rotational degrees of freedom, electronic controllers, and modular holders, see Fig. 1. For

the experiments reported here it was only necessary to use the SWLI, the tip-tilt platform, and a holder.

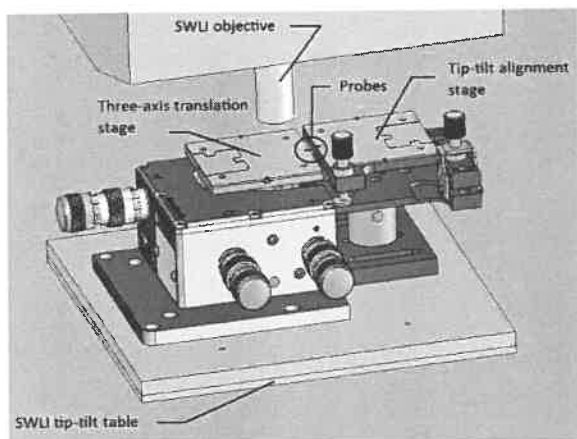


FIGURE 1. The entire setup consists of the SWLI, a translational stage, and a tip-tilt stage enabling cantilever-on-cantilever and cantilever-on-surface bending experiments. Only the tip-tilt stage on the right side was used here.

Equation 3 was evaluated by loading a macroscopic cantilever with known weights and imaging a section of the beam. The cantilever was diamond cut from a $\langle 111 \rangle$ silicon wafer at the flat and had design dimensions of 1 mm wide, 5 mm long, and 0.3 mm thick. After fabrication, the measured dimensions were 0.941 ± 0.004 mm average width and 4.820 ± 0.004 mm length. The elastic modulus for the $\langle 111 \rangle$ direction was specified as 188 GPa and the thickness as 0.3 mm by the wafer manufacturer; see Fig. 2.

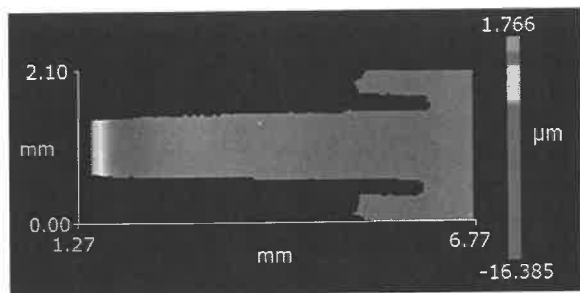


FIGURE 2. SWLI image of the fabricated cantilever. The enlarged field of view was enabled through stitching multiple images together. The width of the beam varies slightly along the length. Roll-off at the free end is due to wafer polishing around the wafer periphery.

The cantilever base was fixed to a holder using Crystalbond™ heat-activated adhesive. The holder was then bolted to the tip-tilt platform. To enable cantilever loading, each weight was attached to its own tether and then weighed using an A&D FX-1200i precision balance (10 mg resolution); see Fig. 3. The cantilever with weights attached is displayed in Fig. 4.

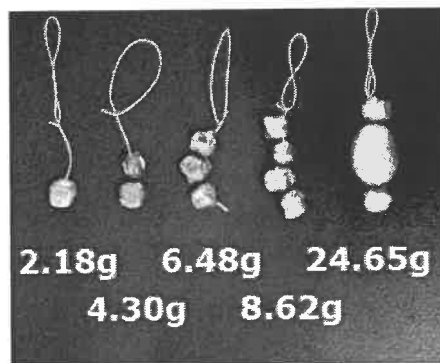


FIGURE 3. Weight set used for loading experiments

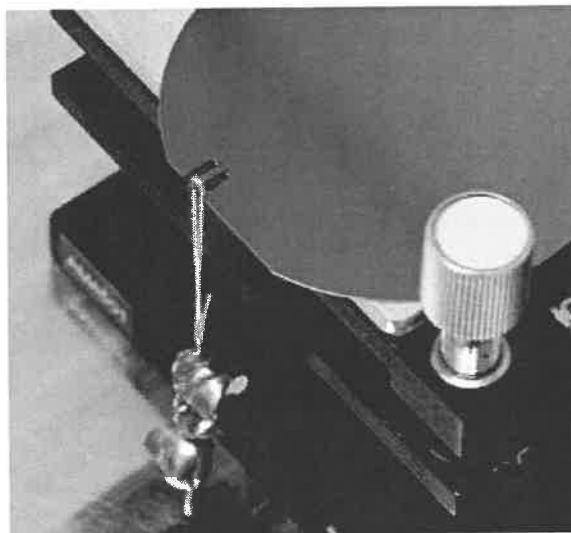


FIGURE 4. The cantilever attached to the tip-tilt stage. The beam is loaded near the free end. Measurements are valid at any section between the base and the load point.

The procedure for obtaining the multipoint measurement began with alignment of the cantilever base to the SWLI optical axis (the optical axis was treated as aligned with gravity although the degree of alignment was not determined). To perform this alignment, a 5x objective with 0.5x system magnification was applied to give a 2.82 mm by 2.12 mm field of

view and 4.4 $\mu\text{m}/\text{pixel}$ lateral resolution (the same settings were used in the force measurements). The cantilever was then laterally positioned under the objective using motorized stages. The cantilever was longer than the field of view so only a section of the cantilever was measured; stitching was not used for the force measurements. Then, the measurement sequence for each weight was:

1. attach the weight to the cantilever
2. complete a SWLI measurement
3. remove the weight
4. complete a SWLI measurement.

The two SWLI measurements were differenced to isolate the deflection caused by loading. This was required for two reasons. First, gravity loading is not included in Eq. 4 so gravity-induced bending must be removed. Second, no beam has a perfectly flat surface so the differencing step eliminates the surface bias. However, differencing requires lateral alignment between images. Therefore, the weight was applied first since it took the most time to set up. This reduced the time between step 2 and 4 to a few seconds, which mitigated the effect of lateral drift due to stage settling and thermal effects.

RESULTS

To obtain a beam deflection profile, a section was selected through the center of the cantilever. The data was then fit in a least-squares sense to a cubic polynomial to obtain c_3 . Equation 3 is rearranged to assist in analysis:

$$\frac{c_3}{\text{mass}} = \frac{-9.8 \times 10^{-3}}{6E_{\langle 111 \rangle} I} = \text{const}, \quad (5)$$

where mass is measured in grams, $E_{\langle 111 \rangle} = 188$ GPa, $I = 1/12wt^3$ (rectangular cross-section), and $t = 0.3$ mm (nominally), and 9.8 m/s^2 is gravitational acceleration. There was a noticeable taper in the width, w , so the average width of the beam at the measured section was used. Equation 5 was evaluated as follows: 1) the left side was calculated using the measured values of c_3 and mass; and 2) the right side was calculated based on the beam's physical description. The two values were then compared. Ideally, they would agree within the measurement uncertainty. Upon first evaluation, there was mismatch in the results. However, measuring beam thickness directly to replace the manufacturer's specified value resolved the

mismatch. This was accomplished by placing a glass surface underneath the cantilever and using the SWLI to measure the distance between the top of the glass surface to the top of the cantilever. Thickness was measured to be 0.305 mm and the results are plotted in Fig. 5.

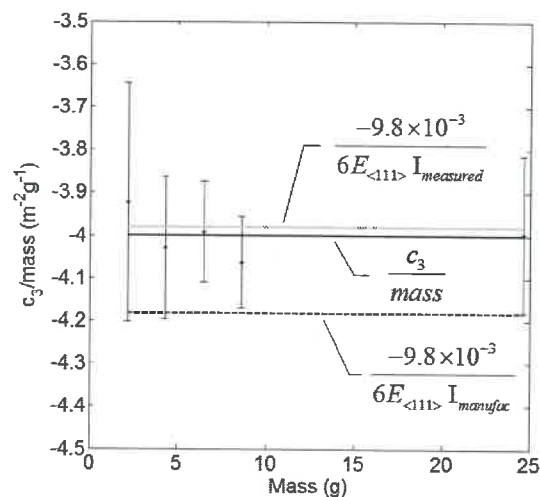


FIGURE 5. Three estimates of the beam constant. Data is reported with error estimates based on 95% confidence intervals. I_{manufac} was evaluated using the manufacturer's specified value of $t = 0.3$ mm; I_{measured} was calculated using the measured value of $t = 0.305$ mm.

DISCUSSION

The mean c_3/mass (left hand side of Eq. 5) value was compared to the constant evaluated from beam properties (right hand side of Eq. 5) and found to have a difference of less than 1% showing good agreement. The repeatability of c_3 was also investigated. Multiple sections of the cantilever were selected from a single image and used to calculate c_3 ; see Fig. 6. The standard deviation was found to be 0.5% of the measurement result. The deviation is an indicator of the SWLI measurement repeatability. Next, the cantilever was loaded and 11 measurements were completed before removing the load. Only the center section was taken from each measurement and used to calculate c_3 ; see Fig. 7. The standard deviation was found to be 1.5% of the measurement result. The deviation is an indicator of the environmental noise. For example, it was observed that the cantilever oscillated continuously while loaded. Therefore, each measurement was a snapshot of the cantilever at different points in the oscillation which

contributed to measurement-to-measurement deviations.

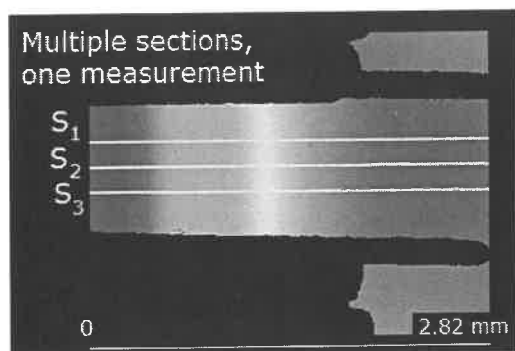


FIGURE 6. Sections were taken proximally to the center.

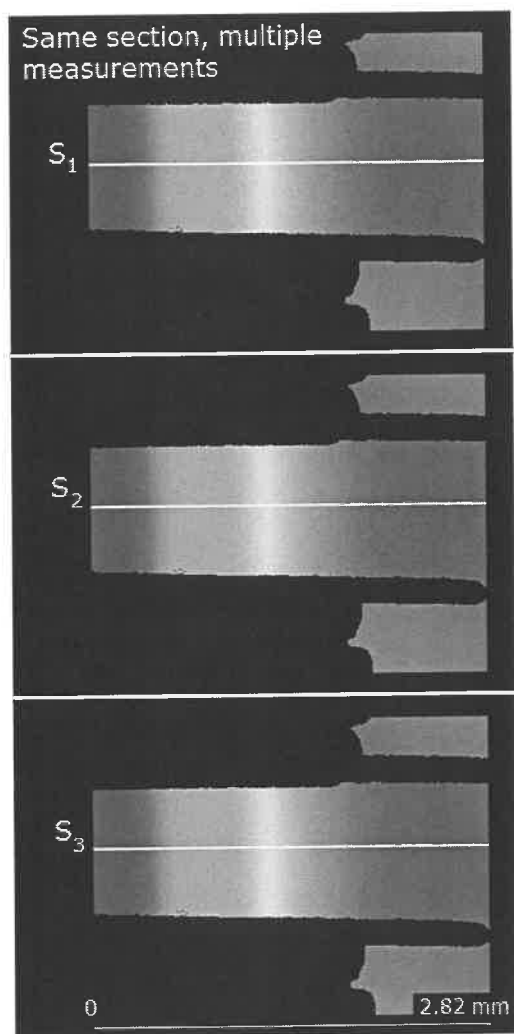


FIGURE 7. Sections were taken from the center.

To test the measurement location independence of c_3 , a measurement was taken near the free end and compared to one taken near the base. The values of c_3 differed due to the beam's variable width. However, the force prediction was accurate provided the correct average width was used. The result of this experiment is illustrated by the ratio of two measurements of the same force at different locations along the beam:

$$\frac{(c_3 w_{avg})_{base}}{(c_3 w_{avg})_{free}} = 1, \quad (6)$$

where the subscript base refers to a measurement made at the base, the subscript free refers to a measurement made near the free end, and the width is the average in each respective view (all other terms cancel each other). Nominally the result should be equal to 1. For the experimental result, the ratio was determined to be 0.996 which is within the measurement uncertainty of the nominal value of 1.

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INFLUENCE OF ALIGNMENT ERROR BETWEEN TWO LASER BEAMS ON STRAIGHTNESS MEASUREMENT USING IN-SITU DIFFERENTIAL LASER AUTO-COLLIMATION METHOD

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INTRODUCTION

Demands for long mechanical and optical parts which have high straightness or flatness are increasing. Examples are seen in a flat panel for liquid crystal display, a throttle die for thin film production and machine elements of semiconductor equipments. Although the length of the parts exceeds 1 m, the requirement for their figure accuracy is higher than $0.1\mu\text{m}$. However, there is no useful method to evaluate precisely the figure accuracy of the long parts. The reasons are inevitable difficulties in manufacturing long tangible datum and to avoid the effect of gravity and environment. Developing the ultra-precision measuring machine for long parts is an urgent need.

In the previous paper, a new concept of straightness measuring machine based on differential laser auto-collimation (DLAC) is proposed. By DLAC method, the difference in inclination angle is successively measured between two probing points along a straight line on a surface to be measured as a discrete data at every probing distance. A parallelism error of two laser beams larger than the resolution of angle sensors can be compensated in-situ. In this paper, to design a suitable angle sensor, influence of alignment error between two laser beams on the measuring accuracy was investigated. Moreover, the influence of alignment error in DLAC was compared theoretically with that of zero setting error between displacement sensors in successive three point method to measuring accuracy.

IN-SITU SELF CALIBRATION LASER DIFFERENTIAL AUTO-COLLIMATION

A schematic drawing of in-situ self calibration DLAC is shown in Figure 1. To eliminate the alignment error of laser beams and angle detectors, the system consists of two angle

detectors (AD_a , AD_b), which simultaneously detect the angles in inclination of the surface at two measuring points on the surface to be measured, two laser diodes (LD_a , LD_b) and a rotary table with two polarization beam-splitters (PBS_a , PBS_b) and two quarter-wave plates (QWP_c , QWP_d). In Figure 1, the inclination angles of the reflected laser beam $T(x_1)$ and $T(x_2)$ from the points x_1 and x_2 on the surface $S(x)$ are simultaneously detected by the angle detectors AD_a and AD_b , respectively. d is the distance between x_1 and x_2 (probing distance). The new DLAC method proposed can detect the difference in inclination, $T(x_1) - T(x_2)$, by eliminating not only error motion in the scanning but also the alignment errors of the devices in the system.

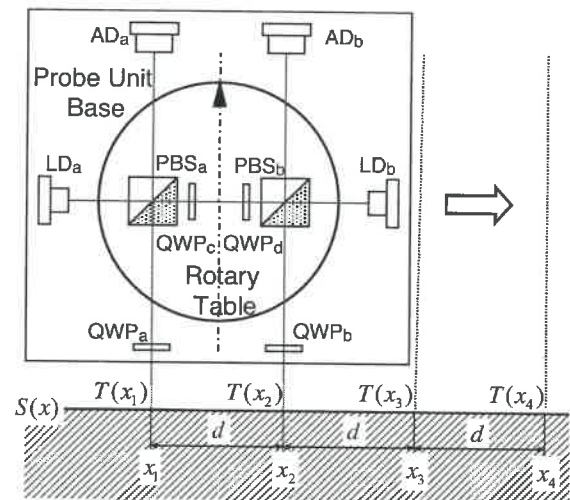


FIGURE 1. In-situ self calibration Differential Laser Auto-Collimation

INFLUENCE OF ALIGNMENT ERROR BETWEEN TWO LASER BEAMS TO FIGURE ESTIMATION

The differential output signal measured by in-situ DLAC is expressed by Eq(1).

$$Ds(x_i) = T(x_{i-1}) - T(x_i) \quad (1)$$

Assuming that both of the surface inclination and offset distance of the probes at $x_i=0$ are zero, the inclination angle at x_i is expressed by Eq(2).

$$T(x_i) = -\sum_{i=1}^i Ds(x_{i-1}) \quad (2)$$

The estimated figure profile $f(x)$, which approximately represents the figure profile, is calculated from $T(x_i)$ using Eq(3). (Fig.2)

$$f(x_i + \frac{d}{2}) = d \sum_{i=1}^i T(x_i) \quad (3)$$

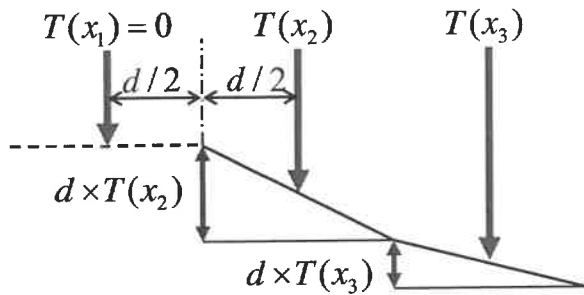


FIGURE 2. The figure profile $f(x)$ estimated from $T(x_i)$

If an alignment error, Ae , is included between two laser beams, Eqs. (1),(2), and (3) should be changed to Eqs. (4), (5), and (6).

$$Ds(x_i) = T(x_{i-1}) - T(x_i) - Ae \quad (4)$$

$$T(x_i) = -\sum_{i=1}^i [Ds(x_{i-1}) + Ae] \\ = -\sum_{i=1}^i Ds(x_{i-1}) - iAe \quad (5)$$

$$f(x_i + \frac{d}{2}) = d \sum_{i=1}^i T(x_i) \\ = -d \sum_{i=1}^i \left[\sum_{i=1}^i Ds(x_{i-1}) - iAe \right] \quad (6)$$

Assuming that the total number of probing points is n , the estimated height at the end of measuring distance, $f(x_{n-1} + \frac{d}{2})$, is expressed by Eq(7)

$$f(x_{n-1} + \frac{d}{2}) = -d \sum_{i=1}^{n-1} \sum_{i=1}^{n-1} Ds(x_{i-1}) - d \sum_{i=1}^{n-1} iAe \\ = -d \sum_{i=1}^{n-1} \sum_{i=1}^{n-1} Ds(x_{i-1}) - d \frac{n(n-1)}{2} Ae \quad (7)$$

The last term of Eq.(7), $d \frac{n(n-1)}{2} Ae$, means an

accumulation of the alignment error which is a quadratic function of number of measuring points. Figure 3 shows the difference between the estimated profiles without (black line) and with (green line) the alignment error, respectively.

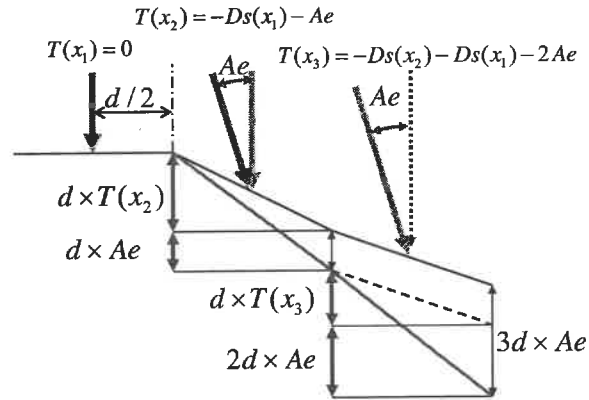


FIGURE 3. The figure profile $f(x)$ estimated from $T(x_i)$ with alignment error

NUMERICAL SIMULATIONS OF FIGURE ESTIMATION

To investigate the influence of the alignment error to estimated profile, numerical simulation was carried out. The figure to be measured was supposed to be a sinusoidal wave. Figure 4 is one of the simulation results when the alignment error was $100\mu\text{rad}$. The probing distance d , the length of measured surface $L=nd$ and the wavelength and amplitude of supposed figure profile are 30mm, 1800mm and 900mm and $2\mu\text{m}$, respectively. A quadratic component is observed in the estimated profile with the alignment error (red line). Figure 5 shows the difference between the estimated figure and true one when the distance between two laser beams was changed from 30mm to 50mm, keeping the alignment error Ae at $1\mu\text{rad}$. The measuring error decreases as the probing distance increases.

Figure 6 shows the influence of the probing distance and alignment error to measuring error. The measuring error increases as the probing distance decreases and the alignment error increases. Figure 7 shows the influence of spatial frequency of the surface profile component. The amplitude of measured profile was not influenced by the spatial frequency.

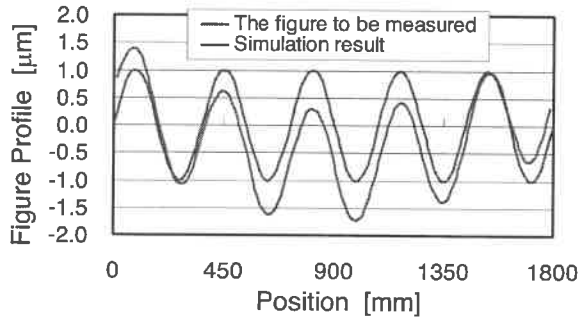


FIGURE 4. The Simulation result ($A_e=100\mu\text{rad}$)

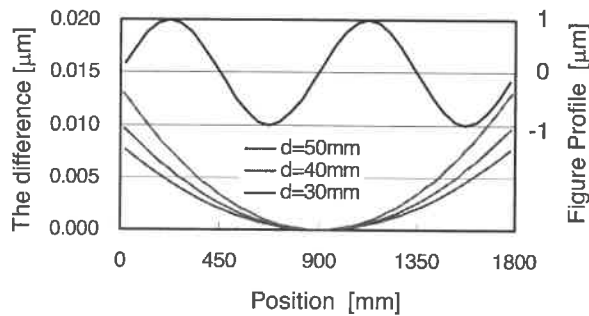


FIGURE 5. The difference between the estimated figure and true one ($A_e=1\mu\text{rad}$)

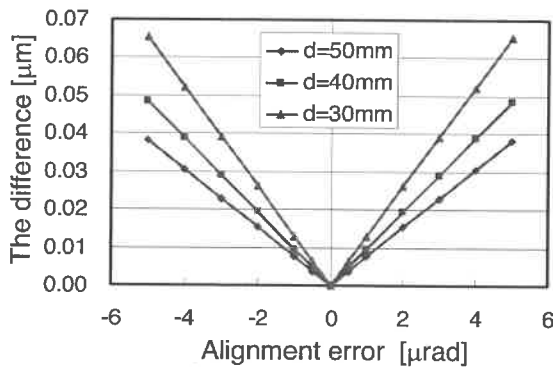


FIGURE 6. The Influence of alignment error

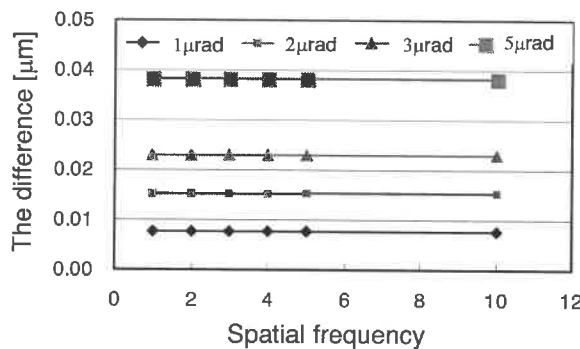


FIGURE 7. The influence of spatial frequency

INFLUENCE OF ALIGNMENT ERROR IN DLAC AND SUCCESSION THREE POINT METHOD

The influence of alignment error in DLAC was compared theoretically with that of zero setting error between displacement sensors in successive three point method to measuring accuracy as shown in Figures 8. The differential output of two displacement sensors corresponds to the inclination angle between the two probing points. Therefore, the relationship between the output signals of the displacement sensors $S_r(x)$, $S(x)$, $S_f(x)$ and the output signals of corresponding angle sensors are expressed Eqs. (8),(9). Both of the differential inclination angle and estimated figure are expressed by Eqs. (10),(11))

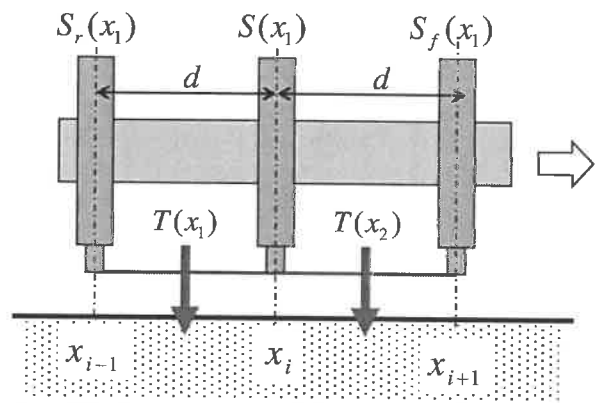


FIGURE 8. Successive three point method

$$T(x_i - d/2) = \frac{S(x_i) - S_r(x_i - d)}{d} \quad (8)$$

$$T(x_{i+1} - d/2) = \frac{S_f(x_i + d) - S(x_i)}{d} \quad (9)$$

$$\begin{aligned} Ds(x_i - d/2) &= \frac{S(x_i) - S_r(x_i - d)}{d} - \frac{S_f(x_i + d) - S(x_i)}{d} \\ &= \frac{2S(x_i) - S_r(x_i - d) - S_f(x_i + d)}{d} \end{aligned} \quad (10)$$

$$f(x_i) = -d \sum_{i=1}^i \sum_{i=1}^i Ds(x_{i-1}) \quad (11)$$

If an zero setting error α is included between the displacement sensors, Eqs. (10) and (11) should be changed to Eqs. (12) and (13). (see Fig.9)

$$Ds(x_i - d/2) = \frac{2S(x_i) - S_r(x_i - d) - S_f(x_i + d)}{d} - \frac{\alpha}{d} \quad (12)$$

$$f(x_i) = d \sum_{i=1}^i T(x_i - d/2)$$

$$= -d \sum_{i=1}^i \left[\sum_{i=1}^i Ds(x_{i-1} - d/2) - i \frac{\alpha}{d} \right] \quad (13)$$

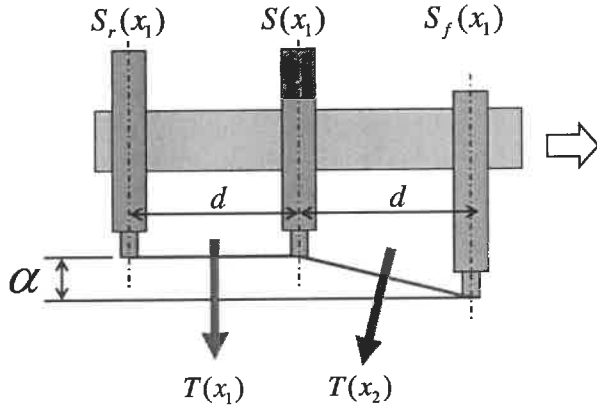


FIGURE 9. Successive three point method with zero setting error α

The estimated figure profile $f(x_{n-1})$ is expressed by Eq.(14)

$$f(x_{n-1}) = -d \sum_{i=1}^{n-1} \sum_{i=1}^{n-1} Ds(x_{i-1} - d/2) - d \sum_{i=1}^{n-1} i \frac{\alpha}{d}$$

$$= -d \sum_{i=1}^{n-1} \sum_{i=1}^{n-1} Ds(x_{i-1} - d/2) - d \frac{n(n-1)}{2} \frac{\alpha}{d}$$

$$= -d \sum_{i=1}^{n-1} \sum_{i=1}^{n-1} Ds(x_{i-1} - d/2) - \frac{n(n-1)}{2} \alpha \quad (14)$$

The last term $\frac{n(n-1)}{2} \alpha$ means an accumulation of the zero setting error which is a quadratic function of number of measuring points.

Both of the term $d \frac{n(n-1)}{2} Ae$ in Eq.(7) and the term $\frac{n(n-1)}{2} \alpha$ in Eq.(14) seems to be a quadratic function of number of measuring points. However, in DLAC, $d \frac{n(n-1)}{2} Ae$ can be expressed as $\frac{n-1}{2} L$ because $L = n \times d$. This

result means that the amount of accumulated error in DLAC is $1/n$ times smaller than the one in three point method.

SUMMARY

The influence of alignment error between two laser beams in DLAC method on the straightness measurement is investigated

- (1) The accumulated measuring error increases as a quadratic function of number of measuring points.
- (2) The measuring error is not influenced by the amplitude and spatial frequency of figure component.

The influence of alignment error in DLAC is compared to the one of zero setting error of three displacement sensors in successive three point method.

- (1) The accumulated measuring error increases as a quadratic function of number of measuring points.
- (2) The amount of accumulated error in DLAC is $1/n$ times smaller than the one in three point method.

Finally, the effectiveness of the proposed in-situ self calibration DALC is confirmed.

ACKNOWLEDGMENT

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MACHINING PROCESSES FOR PRECISION MOLDS

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INTRODUCTION

The continuing drive towards miniaturisation in a number of industries including semiconductor, optoelectronic and medical is stimulated by requirements which include enhanced functionality, performance and reliability, increased energy efficiency, a cleaner environment, improved healthcare, together with reduced costs etc. [1, 2]. Also, new products for fast-growing industries like IT and electronics are increasingly implemented by using ever smaller and more complex modules. Because of the continuing demand for miniaturisation, complex surface figures with and without precise microstructures are becoming more and more important in technical applications. The high volume production of these parts requires reliable replication techniques by the usage of long term durable molds. Here, the challenge lies in the development of sufficiently precise and accurate, deterministic manufacturing processes for these molds.

There is a large number of machining processes available for making precise mold inserts for e.g. out of metal-based, machinable materials like stainless steel or advanced, difficult-to-cut materials like semiconductors or ceramics. The applicable processes differ in the spectrum of machinable geometry, removal rate, achievable form accuracy, surface roughness and sub-surface integrity. Therefore, hard milling and diamond machining are state-of-the-art machining technologies with high potentials and various opportunities for further process improvements that make these technologies more effective and flexible. Otherwise, precision grinding and polishing are limited in contrast to e.g. diamond machining, but harder materials like ceramics are only machinable by these bounded or loose abrasive precision processes.

HARD MILLING

The micro-cutting of steel with tungsten carbide tools can meet many of the demands of miniaturised components. In contrast to macroscopic milling a homogeneous material behaviour cannot be assumed. Varying grain size and

dislocations in the material are annoying the process stability. A prerequisite for a stable machining process is a homogeneous, hard workpiece material with no internal stresses. These tools are commercially available down to a diameter of 50 μm and are still produced by tool grinding.

Nevertheless, this technology is restricted by material, hardness and aspect ratio. The achievable maximum surface roughness is below 3 μm Sz. But the most important process parameter is the cutting velocity. Depending on the hardness of the material, higher cutting velocities lead to better surface qualities, whereby surface roughness of 200 nm Sq and below can be achieved (cf. Fig. 1).

For wear reduction special coatings of TiAlN are applied. The actual coating thickness is reduced under one micrometer, so the increasing rounding of the cutting edge is not effective.

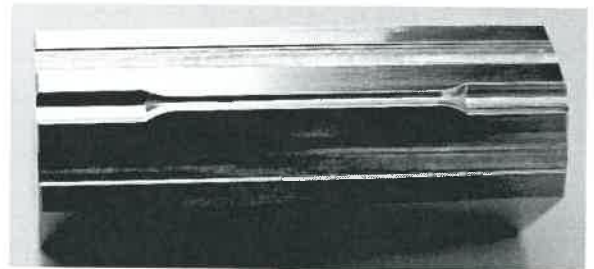


FIGURE 1: Hard milled mold of hardened cold working steel (60 HRC) for micro rotary swaging of thin wires

DIAMOND MACHINING

The largest spectrum of geometries is machinable by diamond turning and milling processes. The processes are adaptable to continuous and structured surfaces, whereby the machining of structured surfaces can be done by circumferential or ball end milling. For machining structures like prisms or spherical lens arrays, diamond tools with high form and angular accuracy are necessary. Moreover, the tools have to be in an accurate position within the tool holder and in relative position within the coordinate system of the machine tool. To

prevent geometrical deviations when machining micro structures, the diamond tool has to be positioned in the machine tool with respect to its coordinate system.

A novel development for the diamond machining of steel molds is the nitriding of the workpiece material. Due to the catastrophic tool wear when diamond turning steel, the spectrum of machinable materials is so far limited. Thus, the amorphous electroless nickel coating is the only diamond machinable hard material (approx. 550HV). This is state of the art for making mold inserts for injection molding, but this material suffers from its low hardness and low temperature stability above approx. 200°C, e.g. for hot pressing of glass. Mold materials with sufficient properties are up to now machinable by grinding and polishing only, therefore, structured surfaces or cavities with small radii are almost not machinable. However, two alternative processes seem to beat this limitation. The ultrasonic assisted diamond turning, introduced by Moriwaki [3] and revisited by Brinksmeier [4], and the thermo-chemical modification of the mold material by a new nitriding process [5].

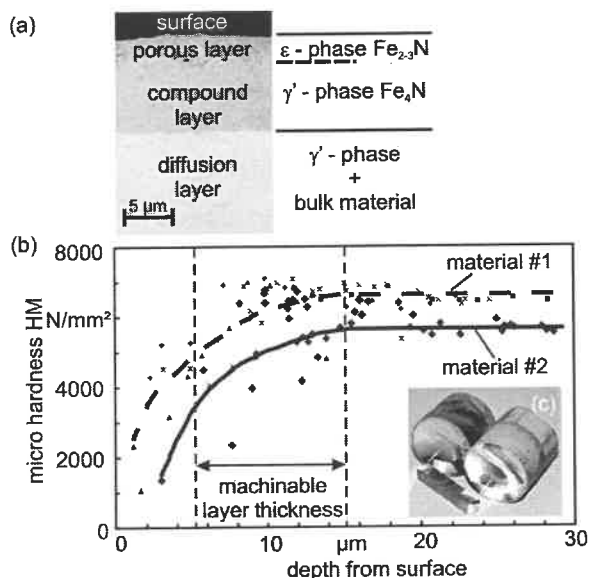


FIGURE 2: (a) Cross section and dominant phases of steel after thermo-chemical treatment; (b) Depth profile of micro hardness for two workpiece material after thermo-chemical modification; (c) diamond turned aspherical hard steel molds turned with one diamond tool

The basic idea of the new method is to avoid chemical reactions between the carbon of the

diamond tool and the iron of the workpiece by establishing a chemical bond between the iron and another chemical element. This was done by a nitriding process, where after the surface zone of the workpiece material (tempering steel 42CrMo4) consist of a layered structure (cf. Fig. 2a). The compound layer is composed of two phases, an ϵ - Fe_2N -phase and a γ' - Fe_4N -phase, where all iron atoms are bonded to nitrogen atoms.

Fig. 2b shows, that the thickness of the compound layer of two used workpieces, a soft, stress relieved steel (340HV) and a hardened steel (730HV), is approx. 15 μm in both cases. Within the compound layer the hardness increases with increasing depth from 2000 N/mm^2 to 6000 N/mm^2 . Therefore, there should be no difference in cutting soft or hardened steel from the chemical point of view. During the development of the nitriding process diamond face turning experiments were performed at sintered high alloy tool steel ($d_w = 22 \text{ mm}$) with a Vickers hardness of approx. 1200HV. Here, surface roughness of better than 10 nm Sa was obtained (cf. Fig. 2c).

Another challenge in mold making is the direct diamond machining of micro cavities. The spectrum of machinable structures by diamond turning and milling is inherently limited by the tool geometry and its intrinsic kinematic. Whereas the machining of retro-reflective triple structures is state of the art in diamond cutting the machining of a more efficient retro-reflective cube corner array cannot be machined because of the need to generate a 3-fold pyramidal cavity. A machining technology that offers the ability to machine sharply ended discontinuous structures in a range between 50 to 500 μm will be a key technology for new applications. With this technology available new optical functions can be implemented in complex optical components leading to a new degree of freedom in optical design.

For meeting this challenge a novel cutting process, diamond micro chiseling (DMC), was being developed [6]. This process enables the generation of discontinuous structures like cube corner arrays with hexagonal aperture (similar to cat's eye of bicycles, cf. Fig. 3). Here, each micro structure has to be cut separately and from several directions (depending on the complexity) and in multiple layers (depending on the aspect ratio). Hence, an ultraprecision machine tool with three linear and two rotational axes is required, which has to be adjusted with a

tolerance below $0.1\ \mu\text{m}$ (resp. 0.1°) in all degrees of freedom. The applied tool is similar to a V-shaped diamond tool and the cutting procedure for a 4-sided pyramidal cavity will take at least 4 individual cuts.

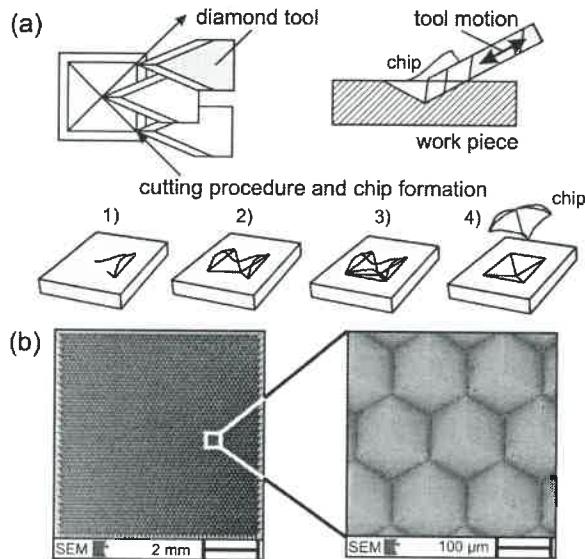


FIGURE 3: (a) Tool motion and cutting process (1-4) for micro cutting of a four sided pyramidal micro cavity; (b) SEM images of four sided pyramidal cube corner array manufactured by diamond micro chiseling in OFHC-copper

PRECISION GRINDING

Precision grinding technologies have to be developed in order to get optical or near optical surface quality when machining ceramic mold materials for mass replication of optical elements. For e.g. cemented carbide composites are widely used in industrial and research applications as die material in glass moulding processes due to their high hardness and low coefficients of thermal expansion, which ensures to maintain the mold shape at the required pressing temperatures. Otherwise, due to the high hardness and brittle behaviour of cemented carbide materials precise grinding processes with fine-grained diamond abrasive tools are essential [7].

Conventional cemented carbides are characterized by a cobalt proportional weight of 5% to 20% and a Vickers hardness of $HV_{10} < 2200$. In contrast, suitable die materials are nano-crystalline binderless tungsten carbides with a grain size below $200\ \text{nm}$, having less than 0.3% cobalt and a hardness of $HV_{10} > 2800$. The binderless type provides the chemical and abrasive wear-resistance of the mould material against the melted glass.

Fig. 4a shows tungsten carbide molds with ground surfaces in pre-machined and precision ground quality out of an optimized precision grinding process with a pin-type, fine-grained, resin-bonded diamond grinding wheel. For the precision grinding process the following parameters have been used: cutting velocity $v_c = 2\ \text{m/s}$, depth of cut $d_c = 0.2\ \mu\text{m}$, tangential infeed velocity $v_{ft} = 0.1\ \text{mm/min}$ and 5%-emulsion as cooling fluid. White-light interferometry and 3D-profilometry of the precision ground surfaces lead to a resulting surface roughness below $4\ \text{nm Sa}$ and a figure accuracy below $150\ \text{nm PV}$ (cf. Fig. 4b).

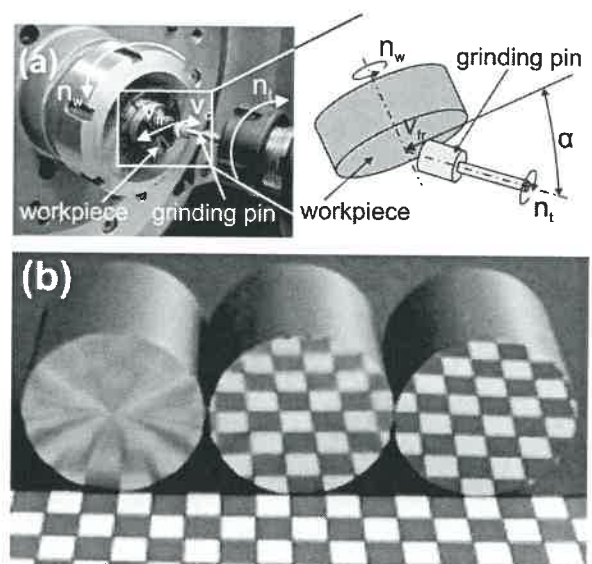


FIGURE 4: (a) Precision grinding of a tungsten carbide mold insert with a pin-type diamond grinding wheel; (b) different ground mold inserts after conventional pre-machining (left), precision rough (centre) and fine grinding (right)

POLISHING

In some cases the surface quality of precision ground molds with structured elements super imposed onto envelope surfaces like plane, spherical, aspheric as well as free formed surfaces is not sufficient for optical applications. Therefore, deterministic polishing processes to improve surface roughness of structured surfaces are necessary. However, since polishing of aspheres, free-forms and structures is a point polishing process, the challenge of the polishing process is to maintain the form accuracy out of the precision grinding process during the polishing process. Several approaches for point polishing of micro and optical components are known [8].

Besides the continuous optics, structured optics with V-shaped or cylindrical grooves with structure size less than 1 mm are increasingly required. These kinds of structures cannot be polished using state of the art polishing technology. A new structure polishing technology to meeting these demands has been introduced in [9]. By using conical pin type tools or conical polishing pads cylindrical and V-shaped grooves with structure size less than 1 mm could be polished after grinding (cf. Fig 5). The goal is, to decrease surface roughness without loss of form accuracy.

Polishing of steel or ceramics is still a process that bases on the experience of the operator of the polishing machine tool. Here, detailed knowledge of the material removal process is necessary for finding optimal process parameters. Moreover, the demands on the form accuracy of polished steel and ceramic molds are almost below one micrometer with aspherical or free-form shapes. Therefore, the necessary step to a deterministic polishing process needs a fundamental knowledge about mechanical and chemical interaction between workpiece and tool during the polishing process.

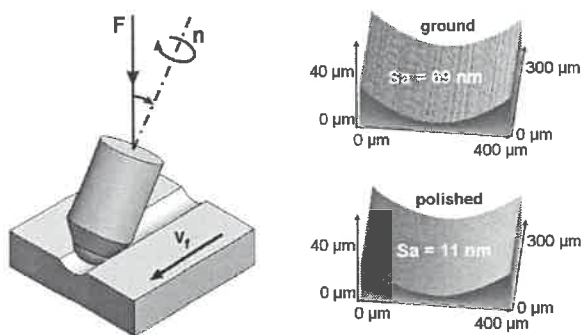


FIGURE 5: Structure polishing process for polishing a cylindrical groove using a plastic polishing pad (right); White light interferometric images of cylindrical grooves in tungsten carbide after grinding and additional polishing (left).

SUMMARY

The mass production of high precision molds with aspheric, structured or free-form shape is still a great challenge, whereas, the number of optics, its complexity and its quality needed increases year by year. Innovative fast, precise and deterministic machining and measuring technologies are required to meet this challenge and the integration of measuring techniques is required. The described machining processes for the manufacturing of mold inserts for injection and glass molding processes, hard

milling and diamond machining as well as precision grinding and polishing, represents possible manufacturing technologies to fulfill the above mentioned requirements.

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METALLOGRAPHIC STRUCTURE EFFECTS ON THE MACHINED SURFACE INTEGRITY IN MICRO-MACHINING WITH Al 6061-T6

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ABSTRACT

A study was carried out to investigate the machined surface integrity of Al 6061-T6 by using a single crystalline diamond (SCD) micro-tool with a 5-axis ultraprecision freeform generator. The micro-size tools used in this study which have a cutting contact length of around 30 μm on the primary clearance face were fabricated by a focused ion beam (FIB). It was found that changing the crystallographic orientation results in a variation in the machined surface integrity. Hard and brittle particles in the soft alloy matrix play an important role in the generation of the voids on the machined surface. Decreasing the cross-feed rate will help reduce the effects of crystallographic orientation and the influence of hard particles on the machined surface. This study contributes to a better understanding of the physics of mechanical machining with micro-tools.

Key words: Micro-size diamond tool, micro-machining, machined surface finish, ductile-regime machining.

INTRODUCTION

Most engineering materials are heterogeneous, made up of more than one phase, with different mechanical properties for each phase. This is because each grain in a polycrystalline material has its own unique crystal orientation and size. Since the grain size of the work material is often of the same order of magnitude as the cutter size, the crystal orientation will strongly affect the overall machining properties [1]. Al 6061-T6 is widely used for optical components since optical grade surfaces can be produced through single point diamond turning. However, the defects generated on the machined surface, including the sub-surface damage, caused by the micro-machining process may seriously affect the functionality of the manufactured micro-parts or

features. So far there have been relatively few reported studies on the machining of aluminium alloys using micro-size SCD tools. The current paper investigates the surface integrity on Aluminium alloys after micro-machining.

EXPERIMENTAL SETUP

Experiments were conducted on a Moore Nanotech 350 5-axis ultraprecision freeform generator. A dual beam FIB system which integrates ion and electron beams for FIB and Scanning Electron Microscope (SEM) functions was used to fabricate the micro-size SCD tools and observe the machined workpieces. The SCD micro-tools used in the study were fabricated by a FIB [2] as shown in Fig. 1 (a). The cutting contact length of the micro-size tool is around 30 μm on the primary clearance face. The rake angle of the micro-size tool is around -1.5° ; the primary and side clearance angles are 7° . The cutting edge radius ranges from 15 to 18 nm as shown in Fig. 1 (b). The machining experiments were carried out on the top surface of the Al 6061-T6 alloy which was previously shaped by a conventional SCD tool with a nose radius of 1.0 mm. The micro-size tools were carefully aligned before performing the machining experiments [3]. The composition and properties of the Al 6061-T6 alloy are shown in Tables 1 and 2, respectively.

RESULTS AND ANALYSES

Fig. 2 (a) shows the micro-groove machined with a micro-size tool at a machining depth of 1.5 μm and a machining speed of 1 mm/min without coolant. The cut was performed with a full top layer being removed with a width equivalent to the cutting contact length on the primary clearance face. Fig. 2 (b) shows the same surface after removing a 0.45 μm layer by FIB sputtering. It was found that different grain orientations resulted in a variation in the machined surface finish. The average surface

finish was below 30nm R_a at grain 1 and greater than 55 nm R_a at grain 2, respectively. Microvoids were found distributing across the machined surface as shown in Fig. 2 (a). Particles were found under the machined surface in the proximity of the voids as shown in Fig. 2 (b). The Nano-indentation study showed that the particle was harder and more brittle than the surrounding matrix material as shown in Fig. 3. The hard and brittle particles were either sheared or dislodged as a whole or broken into smaller pieces when in contact with the micro-tool. This results in the generation of the voids on the machined surface. Fig. 4 schematically shows the ductile-regime machining of brittle materials [4, 5] achieved by single point diamond turning using a round-nosed tool. It exhibits a transition from ductile regime to brittle regime machining. There is a maximum value of feed rate for which ductile-regime machining is possible for a given material and machining parameters. It is reasonable to assume that ductile-regime machining can be implemented into micro-machining of soft matrix alloy materials containing hard and brittle particles. Fig. 5 shows the shaping process with various cross feeds. Fig. 6 (a) and (b) shows the machined surfaces at a machining speed of 1 mm/min and machining depth of 0.8 μ m without coolant, however using different cross-feed rates of 1 μ m/pass and 10 μ m/pass respectively. The

machined surface shown in Fig. 6 (b) was achieved by removing the top layer of the material at a cross-feed rate of 1 μ m/pass, It was found that both the prevalence and size of voids were significantly reduced on the machined surfaces. The machined surface finish was improved to be lower than 10 nm R_a compared with 35 nm R_a obtained with a higher cross-feed of 10 μ m/pass as shown in Fig. 6 (a). Using a reduced cross-feed could help to minimize the effect of grain orientation on the machined surface finish. Reducing or minimising the size of the hard phase would also help to improve the machined surface. A previous study carried out by the authors showed that few defects were detected on the machined surface of NiP, a high purity amorphous material [3].

CONCLUSION

It was found that different grain orientations result in a variation in the machined surface finish. Hard and brittle particles in the soft alloy matrix play an important role in the generation of the voids during micro-machining. Lowering the cross-feed rate can reduce the effect of both hard particles and crystallographic orientation on the machined surface. Micro-machining materials of high purity or those which exhibit minimal hard and brittle particle sizes will produce a higher quality machined surface.

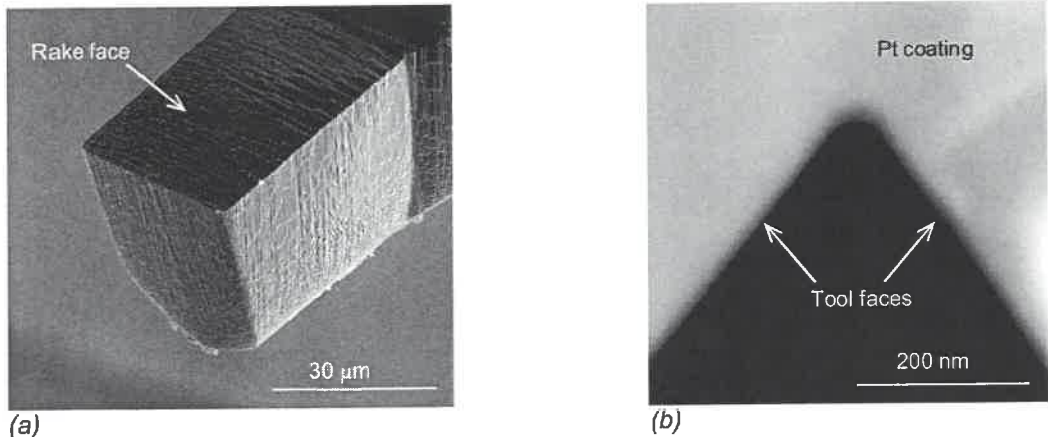


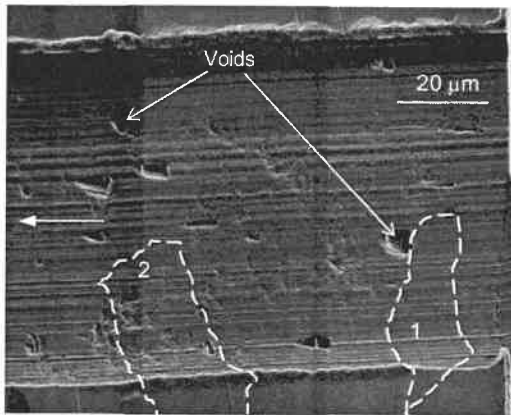
FIGURE 1. View of a SCD micro size tool fabricated by FIB.

TABLE 1. Properties of Al 6061-T6

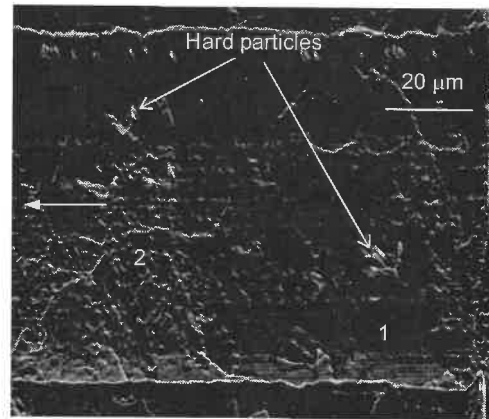
Physical properties		Mechanical properties 20°C			
Density (g/cm ³)	Young's Modulus [Gpa]	Ultimate Tensile Strength (Mpa)	Yield Tensile Strength (Mpa)	Elongation (%)	Hardness (HB)
2.70	70	310	275	12	90

TABLE 2. Typical compositions of Al 6061-T6

Compositions (Weight %)								
Al	Si	Fe	Cu	Mg	Zn	Cr	Mn	Ti
97	0.6	0.7	0.2	1.0	0.1	0.2	0.1	0.1



(a) Before FIB sputtering



(b) After FIB sputtering

FIGURE 2. View of micro-groove machined at a machining depth of $1.5\ \mu\text{m}$ without coolant. Horizontal arrows show the cutting direction.

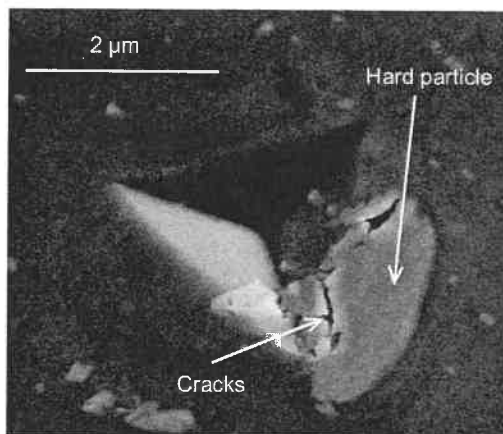


FIGURE 3. Nano-indentation test on Al 6061-T6

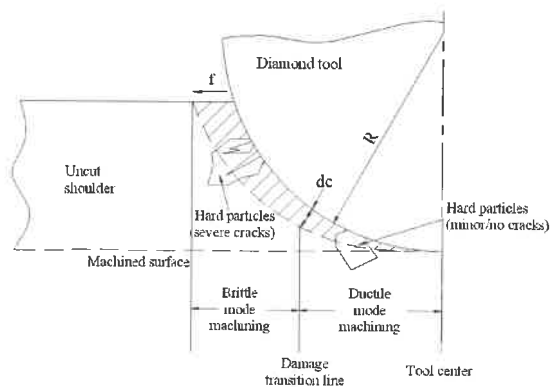


FIGURE 4. Transition from brittle to ductile mode machining of hard particles (adaptation of [4])

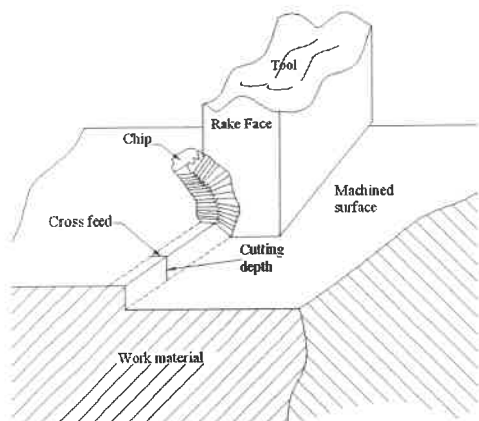
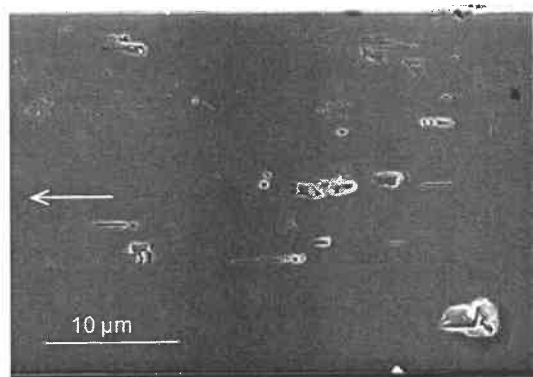
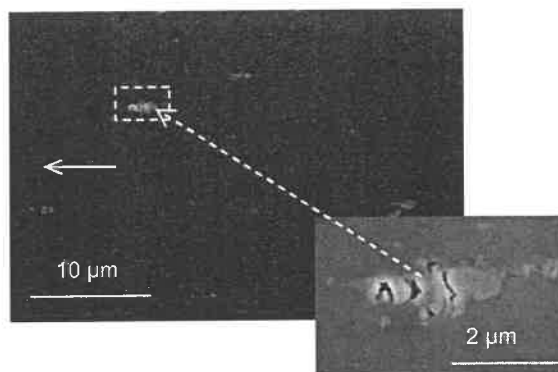


FIGURE 5. Micro-machining with a cross feed.



(a) R_a : <35 nm



(b) R_a : <10 nm

FIGURE 6. Micro-machining with crossing feeds of (a) 10 µm/pass and (b) 1 µm/pass

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THE CRYSTALLOGRAPHIC DIRECTION INFLUENCE ON THE BRITTLE TO DUCTILE TRANSITION IN DIAMOND TURNING OF SILICON CRYSTAL

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INTRODUCTION

The feasibility of cutting brittle materials under conditions that lead to a ductile regime has driven many efforts to the comprehension of the material removal mechanism involved. Consequently, the suppression of the brittle response is a desirable aspect in the machining of semiconductors crystals.

The plastic behavior of silicon during diamond turning has been attributed to a pressure induced structural transformation from diamond cubic structure to a metallic characteristic [1], which had demonstrated that diamond-cubic silicon transforms to a denser metallic β -tin structure at room temperature. In silicon, the critical pressure observed to cause the transformation to a β -tin metallic phase is in the range of 11.3-12.5 GPa and the pressure to cause transformation into the metastable amorphous semiconductor phase, upon unloading, is about 7.5-9.0 GPa [2]. However, the nominal value of Vickers microhardness is different according to the crystallographic direction of the crystal. In (100) monocrystalline silicon, the Vickers microhardness in the [100] and [110] directions are 12.6 GPa and 9.3 GPa, respectively [2].

In the present paper, the dependence of brittle-to-ductile transition on the crystallographic direction in diamond turning of silicon single crystal is investigated. Microindentation and face cutting experiments were conducted in semiconductor single crystals of silicon (001)-oriented in two directions: [100] and [110]. In addition, the cutting conditions were tested in order to study the effect of crystallographic direction on brittle-to-ductile transition when using large feedrates and submicrometer depth of cut.

EXPERIMENTAL DETAILS

Facing cuts were performed on a single crystal Si polished sample. The specimens were in the form of squares (10 x 10 mm) cut from silicon

wafers (100) 1-10 Ω .cm type p ($B = 10^{15}$ - 10^{16} atoms/cm³) of 55 mm diameter and 500 μ m thick. A round nose diamond tool with nose radius of 0.766 millimeters, -5 degree rake angle and a 12 degree clearance angle from Contour Fine Tooling® (UK) was used. A Rank-Pneumo™ (Keene, NH, US) ASG 2500 diamond turning machine was used in the tests. A cutting fluid used was a synthetic water soluble oil with the purpose of cooling. The surface roughness of the diamond turned surfaces was examined by a non-contact type surface measurement system, i.e. Wyko NT 1100 (Veeco Metrology Group). The magnification objectives used in the Wyko NT 1100 was 20X, while the field of view was 232 μ m x 305 μ m respectively. The Vertical-shifting interferometry (VSI) mode was chosen in the experiment. The surface roughness profiles were then plotted by Vision software from Veeco Instruments, Inc. Figure 1 describes the experimental cutting conditions. The crossfeed was inwards.

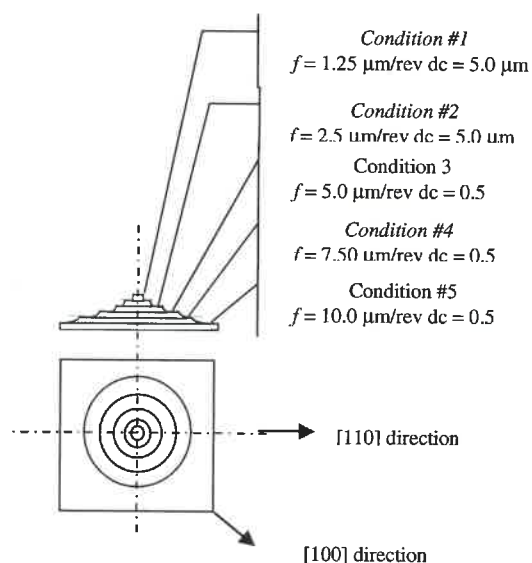


FIGURE 1. Schematic diagram of the cutting conditions applied in the machining tests.

Two cutting conditions were chosen: typical feedrate and depth of cut and large feedrate and sub micrometer depth of cut both aiming at achieving ductile material removal. Consequently, at submicrometer depths of cut using sharp diamond cutting edges, microcracking would be hindered to propagate and higher feedrates could be applied without brittle response.

Cyclic Microindentation tests were carried out using a VMHT MET Leica (Leica Mikrosysteme, GmbH; A-1170, Vienna, Austria) micro indentation machine using a Vickers pyramidal indenter, applying a 150 mN load with dwell time of 15 seconds in cycles of 1, 5, 10 and 15 times and indentation velocity of 60 $\mu\text{m}/\text{sec}$. Microhardness anisotropy measurements were carried out using angular the range of at least 90° , at intervals of 45° .

The micro-Raman spectroscopy study was performed with a conventional T64000 Jobin Yvon spectrometer. The 514.5 nm line of an argon ion laser, focused by an optical microscope on a region of about $1 \mu\text{m}^2$ of the tool surface was used to excite the Raman spectra. The laser power was kept low at about 0.6 mW, in order to avoid heating effects. The laser beam was focused in the sample with 100x lens objective.

RESULTS AND DISCUSSION

Figure 2A) and 2B) show micro Raman scattering of the machined surface in both directions, [001] and [110], respectively. Comparing Figure 2A) and 2B), there is a intensity reduction of the crystalline silicon (c-Si) Raman peak at 521.6 cm^{-1} for direction [001] and [110], respectively and the presence of a broad band centered at about 470 cm^{-1} , can be noticed in both cases, which can be attributed to the optical band of amorphous Si. It is interesting to observe that the increase of intensity can be attributed to the machining in brittle mode with the increase in feedrate in the soft direction, i.e., [110]. In addition, Figures 2A) and 2B) show that crystal peaks shift to the right from the characteristic cubic diamond structure centered in 521.6 cm^{-1} , with the increase in feedrate. This shift is indicative that the machined surface is under compressive residual stress. The value of this stress can be estimated by the formula proposed by Weinstein & Piermarini [3]:

$$\varpi = \varpi_0 + 0.52 P \text{ (Kbar)} \quad (1)$$

where ϖ is the real position of the characteristic crystalline peak, ϖ_0 is the characteristic peak of cubic diamond structure of silicon (521.6 1/cm) and P is the stress within the machined surface in Kbar. The values of the residual stresses estimated by means of this formula are shown in Table 1.

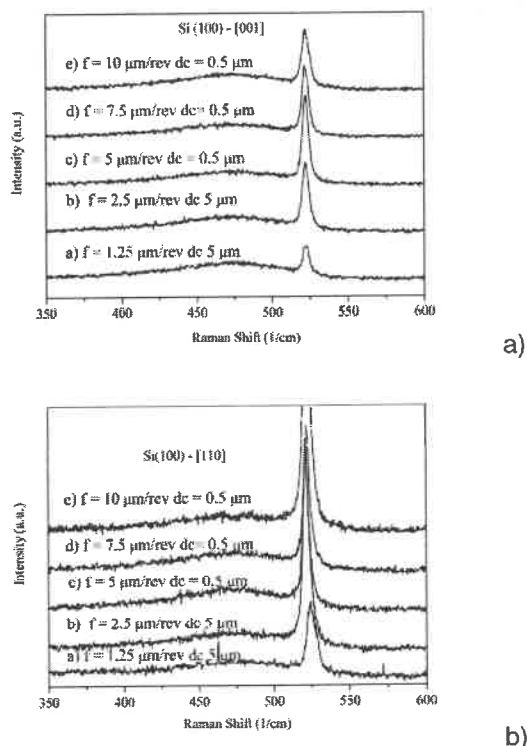


FIGURE 2. Micro Raman spectra from the machined silicon (100) surface; A) [001] direction and, B) [110] direction.

The values obtained show very interesting information: the residual stress increase with the feedrate and no surface stress relief can be detected even with the largest feedrate (10.0 $\mu\text{m}/\text{rev}$) in the [001] direction (Fig. 2A) spectrum [e]). The residual stress in direction [110] is always larger than that in [001] direction. This result corroborates with the fact that the capacity of sustain compressive strain should be larger in the softer direction.

Silicon undergoes phase transformation when the pressure reaches the value within the range of 11.3-12.5 GPa along with the fact that in the presence of shear stress, the diamond structure has been found to become unstable at lower pressures (7-9 GPa) [4]. Based upon this it is possible to assert that it is more difficult to reach the necessary pressure to bring about the phase

transformation in the softer direction, and consequently, to generate ductile response.

Table 1. Estimation of residual stresses on the surface after machining in surface direction [110] and [100] according to peak shift (a-e).

Cut. cond. (spectra)	Peak	Res.	Peak	Res.
	Pos.	stress	Pos.	stress
	(1/cm)	(MPa)	(1/cm)	(MPa)
	[110]	[110]	[100]	[100]
a)	522,84	238,46	522,34	142.31
b)	522,88	246.15	522,24	142.31
c)	523,44	353.85	522,24	123.08
d)	523,33	332.69	522,34	142.31
e)	523,19	305.77	523,14	296.15

At one hand, since [110] direction presents lower hardness, and machining is a dynamic material deformation process; because of this it is more likely that in this direction, both, ductile and brittle modes, take place simultaneously. On the other hand, [001] direction hardness value is within the range of pressure required to silicon undergoes phase transformation and the machining regime to occur totally ductile as it is shown. This is exemplified by Figure 3. Figures 3 a) and 3 c) show that for the [001] direction the uncut shoulder presents ductile regime for both depths of cut, i.e., 5 μm and 0.5 μm . Figures 3 b) and 3 d) show that for the [110] direction the uncut shoulder depicts brittleness for the same depths of cut. In the harder direction, ductile mode was achieved under two opposite cutting conditions, i.e, $f = 2.5 \mu\text{m}/\text{rev}$ $dc = 5 \mu\text{m}$ (Fig. 3a), and $f = 10 \mu\text{m}/\text{rev}$ $dc = 0.5 \mu\text{m}$ (Fig.3c). In the softer direction, a damage free surface finished was only achieved with small feed rate ($f = 2.5 \mu\text{m}/\text{rev}$ $dc = 5 \mu\text{m}$ – Fig.3b). Furthermore, in Fig.3 d), the machined surface presents pits within the cutting groove. The brittle response shown in this direction may be attributed to the fact that the larger difference between transition pressure and hardness can be considered the reason why the microcracks were formed below the cut surface line and on the uncut shoulder as well.

Figure 4 a), spectrum a), shows the Raman spectrum of the machined surface in the [001] direction before microindentation. Figures 4a) spectrum b) up to e) show the sequence of Raman spectra of the cyclically indented machined surface, 1x, 5x, 10x and 15x indentation cycles on the [100] direction. The

intensity of the Raman peaks at 521 $1/\text{cm}$ due to the Si-I formed is increased after the 5th cycle (spectrum c).

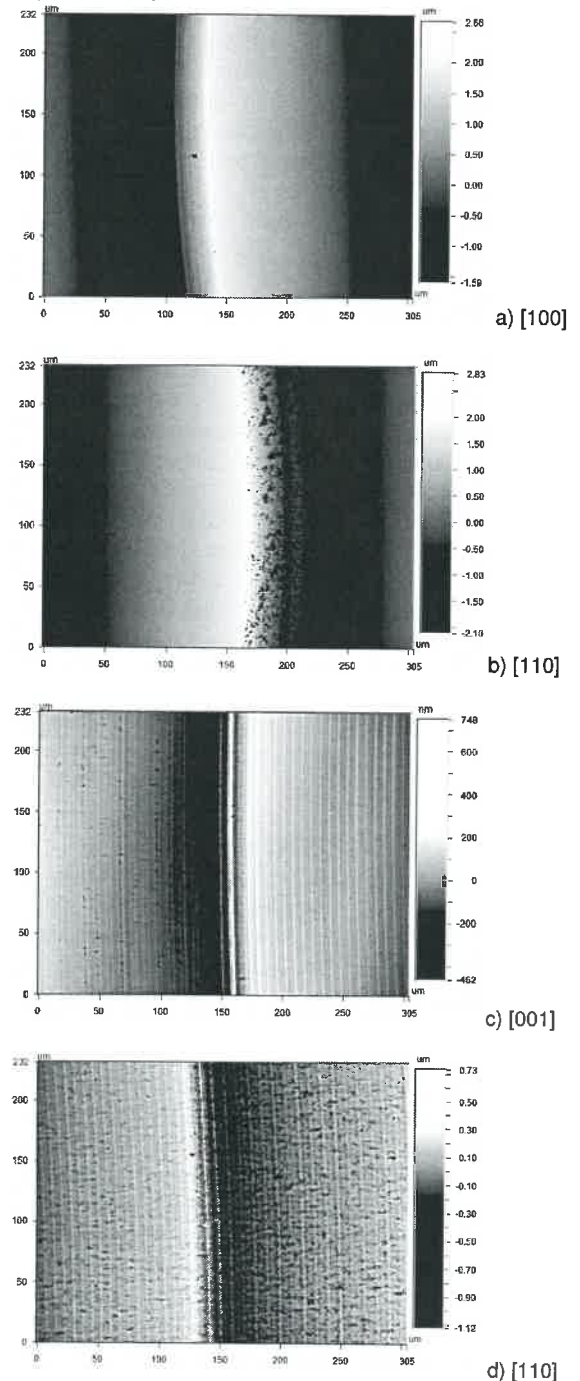


FIGURE 3. Images made by Optical profiler of the uncut shoulder from the machined silicon (100) surface, a) [110] direction and, b) [100] direction. Feedrate used was 2.5 $\mu\text{m}/\text{rev}$ and the nominal depth of cut was kept constant at 5 μm ;c) [110] direction and, d) [100] direction.

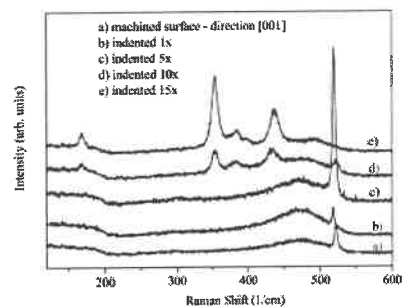
Feedrate used was 10 $\mu\text{m}/\text{rev}$ and the nominal depth of cut was kept constant at 0.5 μm .

In the [100] direction the onset of the formation of new silicon phases were probed after 10 cycles, showing the onset of Si-XII phase formation at 353 $1/\text{cm}$, with the decrease in intensity on the characteristic crystalline peak at 521 cm^{-1} . When the 15th cycle is done (Fig. 4e) the multiple phases present a slight increase in intensity Raman peaks shift. In the Raman spectrum can be identified the following peaks: at 165, 182, 353, 398 $1/\text{cm}$ from Si-XII; 386, 440, 490 $1/\text{cm}$, from Si-III; and at 495 $1/\text{cm}$ from Si-IV [1]. Figure 4 b) shows a sequence of non-indented, 1, 5, 10 and 15 indentation cycles of machined Si in the [110] direction. From the first up to the tenth indentation cycle (spectra b and d) the only effect is the generation of an amorphous phase along with the crystalline phase (a-Si +c-Si), denounced by the broad band at 470 $1/\text{cm}$, showing the peak at 521 cm^{-1} , due to the Si-I, in the diamond cubic structure after the 10th step. The formation of a multiple phase's state with several different structural phases is only produced after the 15th cyclic indentation, as displayed in spectrum e).

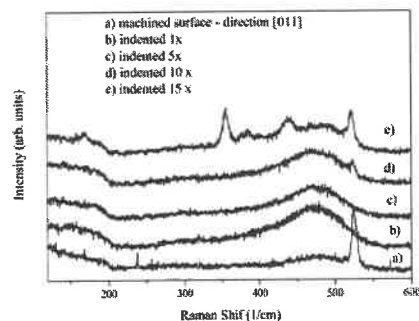
4. FINAL CONSIDERATIONS

Summarizing, we have performed a study on the effect of the crystallographic direction on the generation of brittle and ductile mode material removal in silicon crystal. Residual stresses were estimated by means of the results showed that the [110] direction, which is the softer direction, presented the largest values of residual stresses with maximum 350 MPA. The machined surface presented pits in the [110] direction when submicrometer depth of cut and large feed rates were used. The results for [100] presented the opposite response when compared with the [110] direction in both cases. The difference in the brittle and ductile behavior, for different crystallographic directions, was attributed to the difference between microhardness and transition pressure value to induce phase transformation during machining. The harder direction [100] presents microhardness value approximately the same as the transition pressure value of silicon, which undergoes ductile material removal when phase transformation takes place. Since the softer direction [110] has a lower hardness, the difference between the H_V and transition pressure value will be larger and then the phase transformation will be more difficult to take

place. Consequently, the onset of brittle mode will be more favorable to take place.



a)



b)

FIGURE 4. a) Raman spectra of (001) Si surface successively indented with 150 mN load in the direction [110] a) machined surface, b) Raman spectra of (100) Si successively indented with 150 mN load in the [100] direction.

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ULTRAPRECISION MICRO-MACHINING OF DIFFRACTIVE OPTICS

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ABSTRACT

We have been manufacturing JEM-EUSO lenses. These lenses are manufactured by a large ultraprecision turning machine tool. This machine enables to manufacture optics with ultraprecision, and has been developed with a purpose of machining for large diameter Fresnel lenses. JEM-EUSO optics has two double-sides spherical fresnel lenses and a precision fresnel (diffractive) lens for chromatic aberration correction. Each JEM-EUSO lenses consists of a center part and an outer ring part. The center part is about 1500mm in diameter. JEM-EUSO optics fabrication and application team manufactures three center-lenses of baseline design optics. We describe in this paper the progress of the diffractive lens manufacturing.

1. INTRODUCTION

The JEM-EUSO telescope will be attached to the Japan Experiment Module/ Exposure Facility (JEM/EF) of the International Space Station (ISS). The telescope will observe from 430 km in altitude and enables to cover 400 km in diameter range of an Extensive Air Shower (EAS) made by the highest energy particles in the Earth's atmosphere.

Figure 1 shows an optics module design of the JEM-EUSO telescope. Optics of the telescope is composed of two double-sides spherical fresnel lenses, a precision fresnel (diffractive) lens, and Focal surface detectors. Each Fresnel lens is 2650mm in diameter and collects near ultraviolet photons (330 nm - 400 nm) from $\pm 30^\circ$ field of view with an angular resolution of 0.1° on the focal surface.

Lense1 and Lens3 are curved and apply the double sided Fresnel structure; therefore, lenses realize covering a wide field of view and lighter weight. Fresnel structure and diffractive structure are applied to both sides of Lens2. Role of this surface is to reduce still remained color aberrations by using diffractive effect.

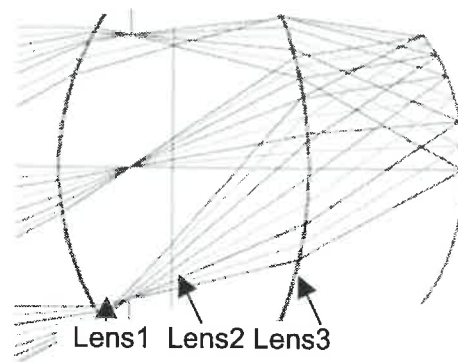


FIGURE 1. Optics design for JEM-EUSO

2. FUNDAMENTAL EXPERIMENT OF TURNING DIFFRACTIVE STRUCTURE

Fundamental experiment of ultraprecision micro-machining diffractive structure of ϕ 200mm has been conducted on an ultra/nanoprecision turning machine as shown in Figure 2. Figure 3 shows the profile of the machined structure. As a result, satisfactory surface roughness of less than 20nm in RMS has been achieved on entire surface of the workpiece as shown in Figure 4. The requested deviation of 10% of 700nm depth grooves has successfully achieved as shown in Figure 5.

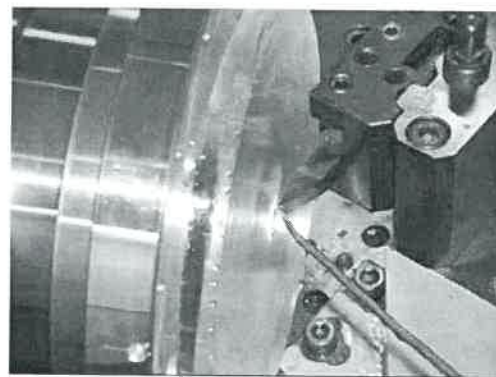


FIGURE 2. View of turning of diffractive structure

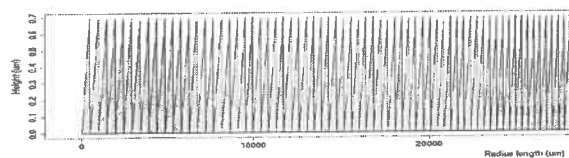
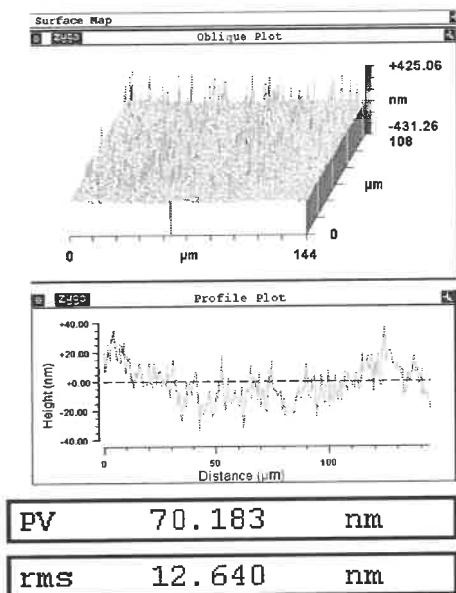
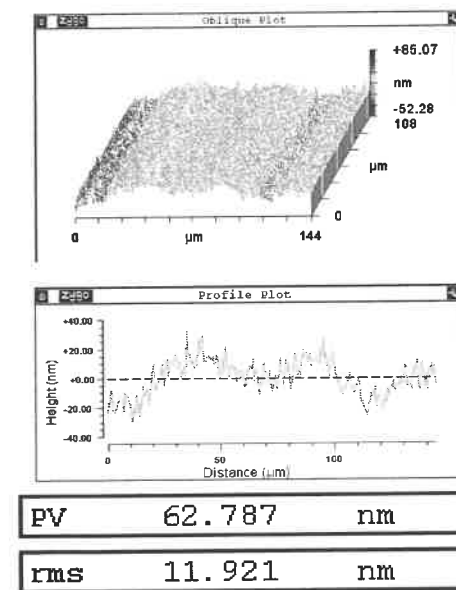


FIGURE 3. Machined structure with 700nm grooves

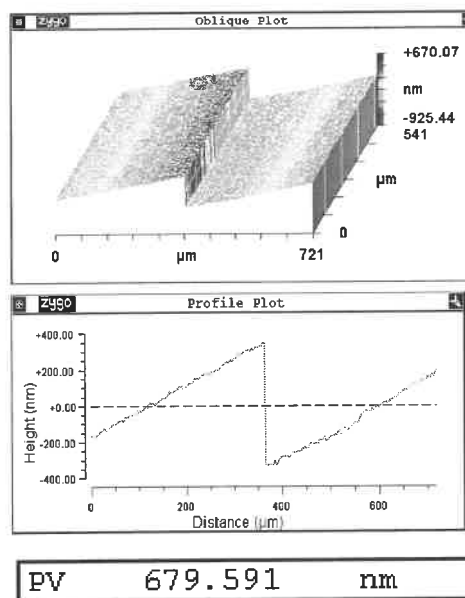


(a) Center area

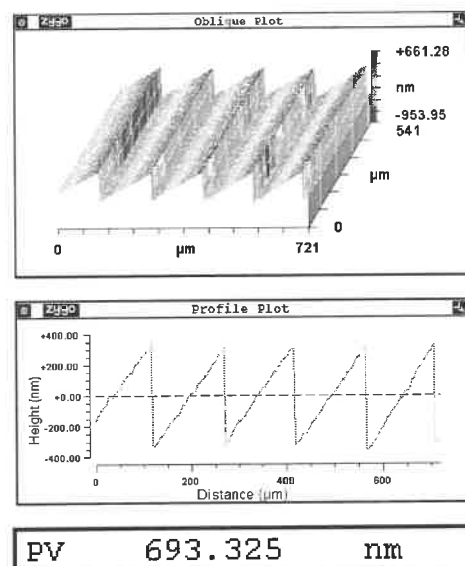


(b) Middle area

FIGURE 4. Surface quality of machined structure



(a) Center area



(b) Middle area

FIGURE 5. 700nm depth groove accuracy

3. LARGE TURNING EXPERIMENT OF DIFFRACTIVE STRUCTURE

A large diffractive structure of 1500mm in diameter has been turned on the large turning machine. Figure 6 shows the view of the turned diffractive structure. Optimization of machining conditions from center, middle to edge area of the large lens has been considered because the peripheral speed

varies very much from the center to edge area on the entire surface of the optics. Figure 7 shows the evaluated accuracy of depth of the diffractive structure. This result showed the accuracy was within 10% deviation of the full depth of the structure. The machined surface quality was also evaluated, and the result showed the surface roughness less than 20nm RMS as requested has successfully achieved as shown in Figure 8.



FIGURE 6. View of turned diffractive structure

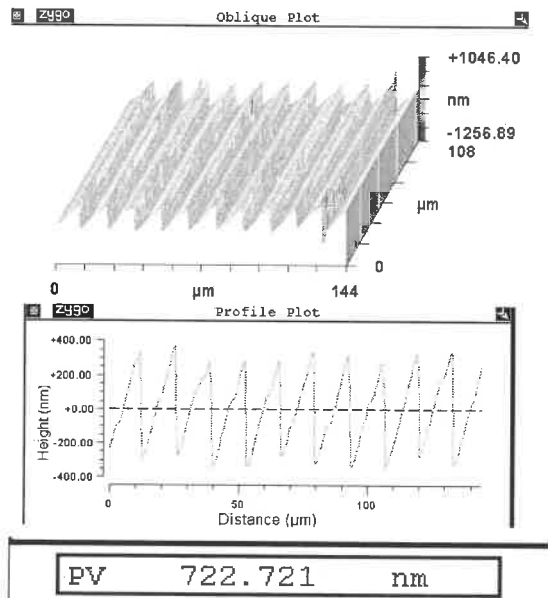


FIGURE 7. Depth accuracy of 700nm diffractive structure

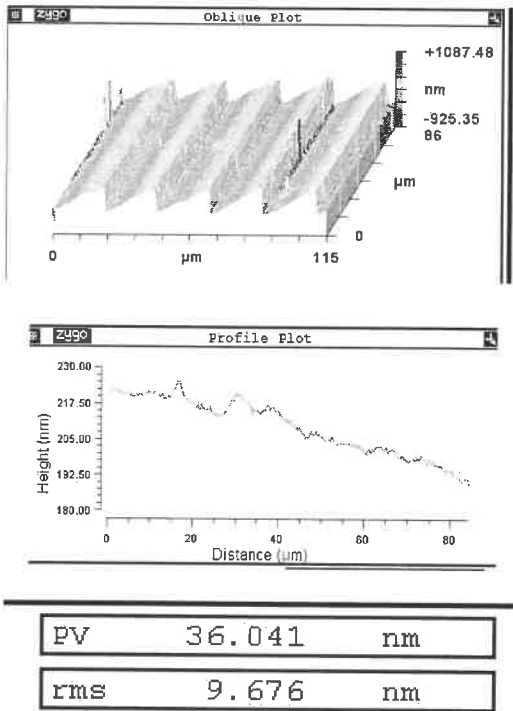


FIGURE 8. Surface quality of turned diffractive structure

5. CONCLUSION

This paper reported our latest achievement on development of diffractive optics for JEM-EUSO. The experimental results showed the required accuracy on both of depth deviation and surface quality had successfully been achieved.

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ULTRA-PRECISION GRINDING OF PMN-PT RELAXOR-BASED FERROELECTRIC SINGLE CRYSTALS

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low-speed (0.001-30 rpm) rotary table. Both spindles have hydrostatic oil bearings which can provide the required high rigidity, rotational accuracy, and vibration damping. The diameter of the grinding spindle is 125 mm. The samples were ground with water as a grinding fluid without recycle. Figure 2 shows the definition of grinding direction on the single crystal sample.

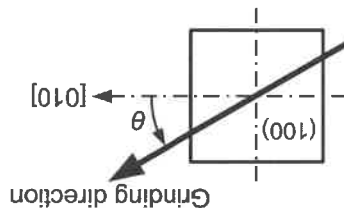


FIGURE 2. Definition of the grinding direction.

The samples were ground with various resinoid-bond diamond cup wheels as shown in Figure 3 and Table 1 at the various grinding conditions using the ultra-precision surface grinder. The outer diameter of the wheel is 125 mm which is the same size with the main spindle, and the width and thickness of the diamond layer are 3 mm. The wheel was dressed by grinding a GC400 stick stone at the conditions of a wheel peripheral speed $V_s=3$ m/s, feed $V_w=0.01$ m/s and depth of cut $a=1$ μ m. All ground surfaces were observed under a Nomarski differential interference contrast microscope of Nikon Corp. to classify the grinding modes, and measured the surface roughness with a three-dimensional optical profiler, NewView 200 of ZYGO Corp.

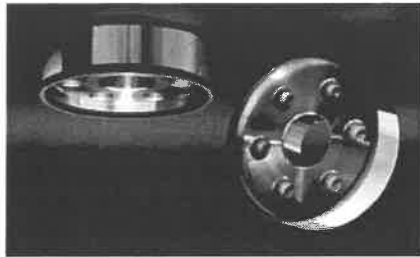


FIGURE 3. Resinoid-bond cup type diamond wheels.

A $(1-x)(\text{Pb}(\text{Mg}_{1/3}\text{Nb}_{2/3})\text{O}_3-x(\text{PbTiO}_3))(\text{PMN-PT})$ single crystal [1] shows high piezoelectric coefficients, extremely large piezoelectric strains and very high electromechanical coupling factors rather than lead zirconate titanate (PZT) which is commonly used in industries. PMN-PT is expected to be used in various sensors of high sensitivity as well as high performance biomedical ultrasound transducers. Some sensors require very smooth and thin plate of PMN-PT single crystal less than 10 μ m without sub-surface damage. This paper deals with the relationship between the ground surface roughness and various grinding conditions for obtaining smooth surfaces as the pre-machining process for the final float polishing process [1].

EXPERIMENTAL PROCEDURE

We used a large PMN-PT piezoelectric single crystal [2] grown by the Bridgman method. The single crystals of (100) face were cut into 4 mm square and 0.7 mm thick by a diamond saw wheel, lapped into parallel and pasted on an alumina plate of 100 mm diameter and 7 mm thick with wax. Figure 1 shows a photograph of the ultra-precision surface grinder [3] having a zero thermal expansion spindle. The machine has a vertical main spindle for a grinding wheel and a

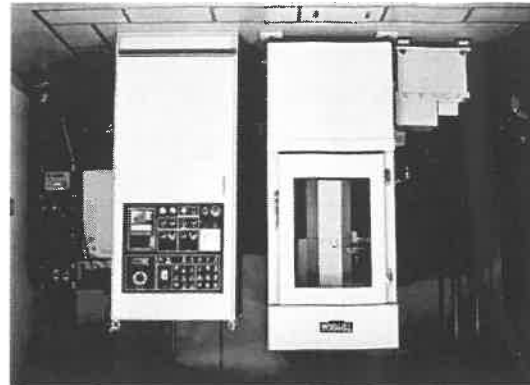


FIGURE 1. Ultra-precision surface grinder.

The Knoop-hardness of PMN-PT (100) single crystal was also measured on the various orientations as shown in Figure 4 with a micro-hardness tester HMV-1 of Shimadzu Corp.

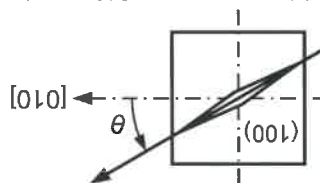


FIGURE 4. Measurement of Knoop hardness on PMN-PT (100) surface as a function of angle θ from [010] direction.

RESULTS AND DISCUSSION

The ground surfaces of brittle materials may be grouped into three classes, namely, fracture mode, ductile mode, and ductile and fracture mode from features observed on the surfaces using Nomarski microscope, as mentioned in previous papers [4-6]. No micro-cracks were found after etching surfaces ground in the ductile mode [4-6].

The PMN-PT (100) samples were ground at the grinding conditions of $V_s=20$ m/s, $a=2$ μ m, feed per wheel revolution $f=1-15$ μ m/rev with various diamond wheels as shown in the Table 1. Then, the ground surfaces were observed with Nomarski microscope and classified into two categories as shown in Figure 5. Figures 5(a) and 5(b) show ground PMN-PT (100) surfaces in the ductile mode and in the ductile and fracture mode, respectively.

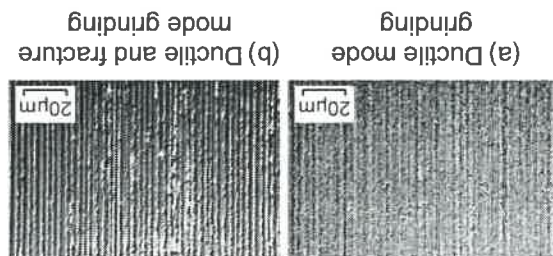


FIGURE 5. Nomarski micrographs of two kinds of ground surfaces on PMN-PT relaxor-based ferroelectric single crystals.

Figure 6 shows the relationship between the grinding modes and grinding conditions, such as feed per wheel revolution and grain size range of diamond wheels. The modes depend on the feed per wheel revolution and grain size of diamond wheels. From this figure, it is clear that the ductile mode grinding can be obtained using the diamond wheels less than 8 μ m in largest diamond grain size, and the diamond wheels having 15 μ m size diamond can not obtain the ductile mode grinding. In the case of using SD2000-J-75-B (grain size range: 5-12 μ m), the feed per wheel revolution affects the grinding mode like Mn-Zn ferrite single crystals, CVD SiC and CaF_2 single crystals [4-6].

After this paragraph we will consider only the ductile mode grinding area for obtaining the super-smooth surfaces.

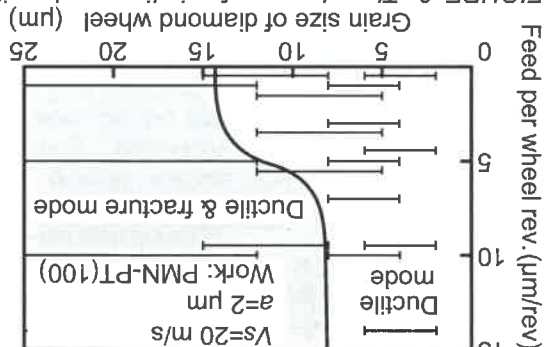


FIGURE 6. The change of grinding mode with feed per wheel revolution and grain size of diamond wheels.

PMN-PT (100) single crystals were ground in various directions to the [010] direction, as shown in Figure 2 at the grinding conditions of $V_s=20$ m/s, $a=2$ μ m, $f=3$ μ m/rev with SD2400-J-75-B wheel. Figure 7 shows the variation of

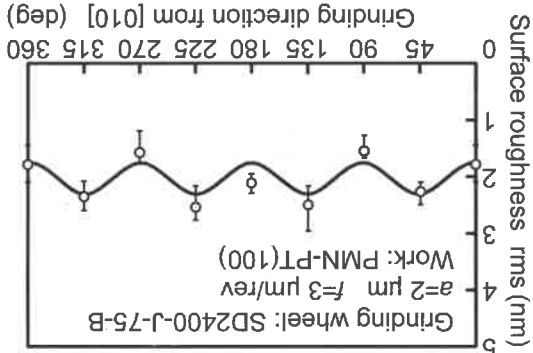


FIGURE 7. Variation of surface roughness of PMN-PT (100) surface ground in various grinding directions from [010] direction.

Figure 10 shows the relationship between the ground surface roughness and average grain size of diamond wheels for the grinding

FIGURE 9. Relationship between surface roughness and feed per wheel revolution.

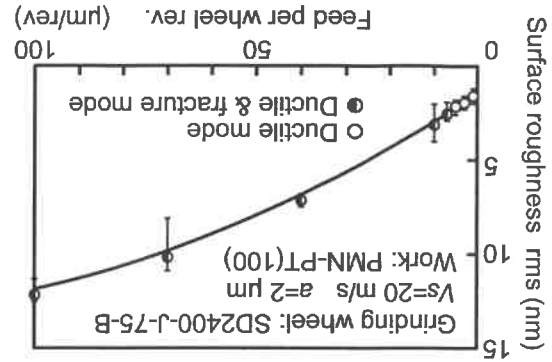
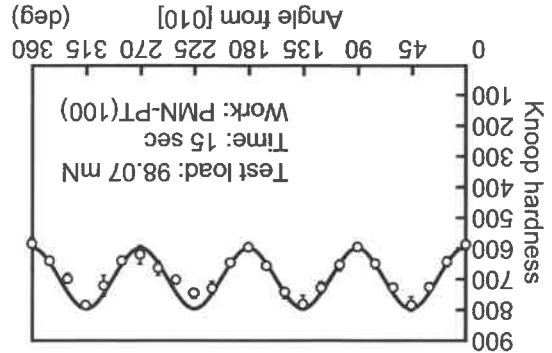


Figure 9 shows the relationship between the ground surface roughness and feed per wheel revolution at the range from 1 to 100 $\mu\text{m/rev}$ for the grinding conditions of $V_s=20\text{ m/s}$, $f=5\text{ }\mu\text{m/rev}$, $a=2\text{ }\mu\text{m}$ and $\theta=0^\circ$ with a SD2400-J-75-B grinding wheel. The ground surface roughness increases with the feed per wheel revolution, and the grinding mode shifts from ductile-mode to ductile & fracture mode over $f=7\text{ }\mu\text{m/rev}$.

FIGURE 8. Relationship between Knoop hardness and orientation of PMN-PT relaxor-based ferroelectric single crystal (100) plane.



surface roughness of PMN-PT (100) single crystal with the grinding direction to the [010] direction in the crystal. It is clear that the ground surface roughness is a function of grinding direction, and the smoothest surface is obtained at the grinding direction $\theta=0^\circ$ and every 90° . Figure 8 shows the relationship between the Knoop hardness and measured direction from [010] direction, as shown in Figure 4. This tendency is very similar to the surface roughness, as shown in Figure 7.

Figure 12 shows the relationship between the ground surface roughness and grinding speed ($V_s=3.3\text{ m/s}$ - 20 m/s) for the grinding conditions of $f=5\text{ }\mu\text{m/rev}$ and $\theta=0^\circ$ with an SD2400-J-75-B diamond wheel. This result comes to a conclusion that the grinding speed does not affect the ground surface roughness.

FIGURE 11. Relationship between ground surface roughness and depth of cut.

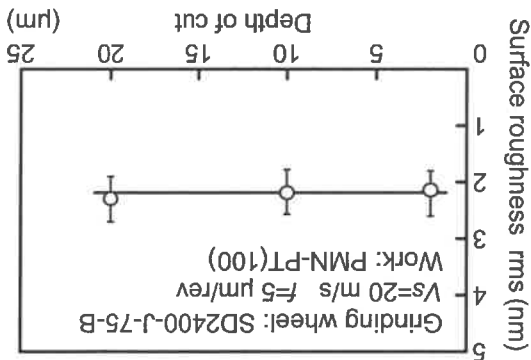
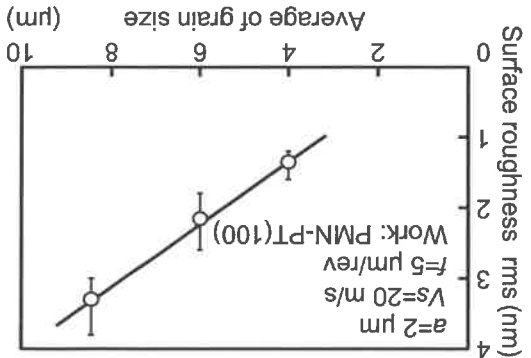


Figure 11 shows the relationship between the ground surface roughness and depth of cut for the grinding conditions of $V_s=20\text{ m/s}$, $f=5\text{ }\mu\text{m/rev}$, and $\theta=0^\circ$ with a SD2400-J-75-B grinding wheel. The surface roughness is constant within a wide range of depth of cut from 1 to 20 μm . All surfaces in this range show ductile-mode grinding and the surface roughness does not depend on the depth of cut. With this result, we are hoping for a marked improvement of productivity with high precision.

FIGURE 10. Relationship between ground surface roughness and average grain size of diamond wheel.



conditions of $V_s=20\text{ m/s}$, $f=5\text{ }\mu\text{m/rev}$, $a=2\text{ }\mu\text{m}$ and $\theta=0^\circ$. The ground surface roughness depends upon the average grain size of diamond wheels.

grinding conditions, particularly grain size of conditions. These grinding modes are related to ductile and fracture modes in this experimental classes, namely, the ductile mode and mixture of 1. The ground surface can be classified into two be drawn from the results of this study.

smooth surfaces. The following conclusions may range of grinding conditions for obtaining ultra-type resin-bonded diamond wheels at the wide low thermal expansion spindle and various cup-precision surface grinder having an extremely-crystal (100) surfaces were ground by an ultra-PMN-PT relaxor-based ferroelectric single

CONCLUSIONS

FIGURE 13. Three-dimensional image of a ground (100) PMN-PT surface measured by a NewView optical profiler.

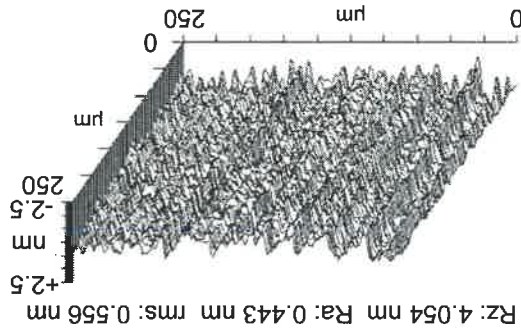
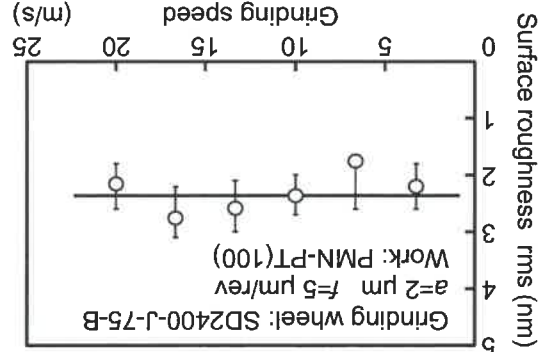


Figure 13 shows the surface profile of PMN-PT (100) sample ground with an SD3000-J-75-B diamond wheel by one pass grinding for the grinding conditions of $V_s=20$ m/s, $f=1$ $\mu\text{m}/\text{rev}$, $a=2$ μm and $\theta=0^\circ$. The surface roughness is 0.44 nm Ra, 4.05 nm Rz and 0.56 nm rms. This surface is smoother than optically-polished one, and has very good edge quality.

FIGURE 12. Relationship between surface roughness and grinding speed.



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ACKNOWLEDGMENTS

3. The surface roughness of 0.556 nm rms, 0.443 nm Ra and 4.05 nm Rz were obtained on ground (100) surface under the grinding conditions of $V_s=20$ m/s, $a=2$ μm , $f=1$ $\mu\text{m}/\text{rev}$ at grinding direction of [010] by one pass grinding with SD3000-J-75-B diamond wheel.

2. Ground surface roughness in the ductile mode grinding depends on the average grain size in a diamond wheel, feed per wheel revolution, crystallographic orientation, and does not depend on the depth of cut and grinding speed.

diamond wheel and feed per wheel revolution.

TECHNOLOGICAL INVESTIGATIONS ON SLOW SLIDE SERVO FOR ULTRA PRECISE FREEFORM MACHINING

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TECHNOLOGICAL RESTRICTIONS OF THE SLOW SLIDE SERVO

In contrast to 2D-programming, the generation of NC-programs for freeforms requires 3D-surface base points which are on an Archimedean spiral according to the tool path. NC-generation can be done with CAM-software like NanoCAM by Moore, Keene or "manually" with math software. For the following investigations described, MATLAB (by The Mathworks, Massachusetts) was used to generate test NC-programs to evaluate slow slide servo behavior.

Number of revolutions depending on the base point distance

Goal of the first described examination was to determine the effect on the machining time of the angular distance between two consecutive base points, and therewith the number of base points per revolution of the C-axis. For this purpose NC-programs were generated with angular distances between 0.05° and 120° and the speed reached during the test runs, was logged. To exclude effects caused by the Z-axis' dynamic, the height distance in Z between all NC-coordinates was zero. The results displayed in Figure 2, show that the maximum speed of the C-axis which is $n_{max} = 2000 \text{ min}^{-1}$, is only reached for angular distances of $W_s = 120^\circ$ which is equivalent to 3 base points per revolution. Increasing the number of base points per revolution - thus reducing the angular distance - results in a decreasing number of revolutions of the C-axis, whereas the correlation is not linear. For an angular distance of $W_s = 1^\circ$ the number of revolutions reached is only $n = 166 \text{ min}^{-1}$.

INTRODUCTION

The demand for free formed surfaces with optical surface quality is increasing. Various applications of such mirrors and respectively lenses are found all over the optical industry and includes for example laser beam shaping. Besides diamond turning with slow slide servo there are other methods available to realize freeform machining, e. g. ultra precision milling, grinding or diamond turning with fast tool servo. Slow slide servo has the advantages of being as accurate, to some extend faster as other technologies, and there is no additional machine equipment needed. Introduced for the machining of non-rotational symmetrical specular surfaces, the slow slide servo technology uses the machine's Z-axis to oscillate the diamond tool synchronized to the motion of the C-axis (controlled main spindle). Figure 1 shows the IWF's ultra precision machine tool equipped with hydrostatic respectively air bearings and linear direct drives respectively torque motors. The machine tool control has an extended look-ahead functionality and is able to perform slow slide servo operations.

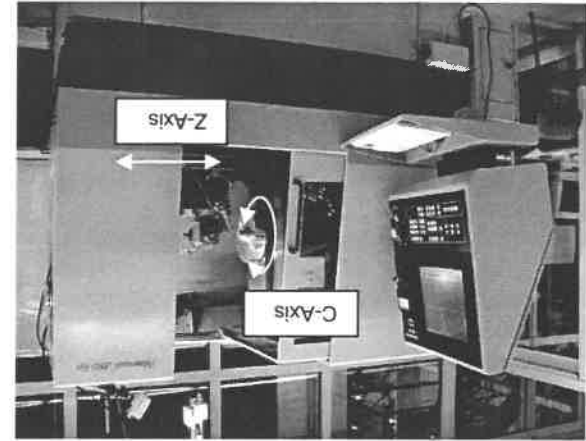


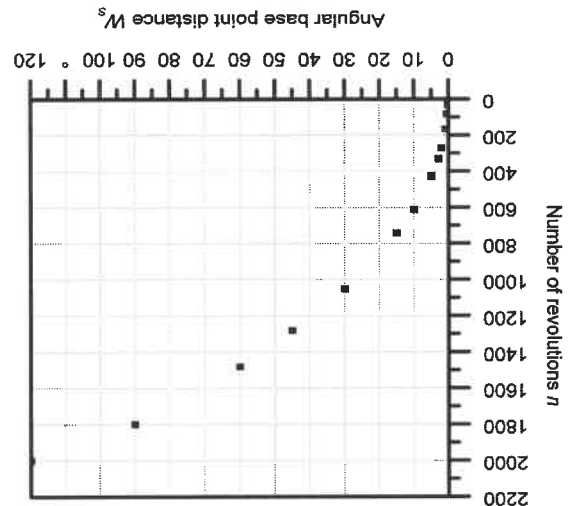
FIGURE 1: Ultra precision machine tool Moore Nanotech@350 FG.

The results of a second examination shall be discussed in the following concerning the positioning behavior of the z-axis. With MATLAB, NC-programs were generated with an angular distance $W_s = 1^\circ$ between the base points and sine wave typed travel paths of different distances for the Z-axis. The travel distances were chosen to be between $l_z = 10 \mu\text{m}$ and $l_z = 15 \text{ mm}$. The number of sine wave typed travels during one revolution of the C-axis was Gold Standard from Renishaw, the travel distance of the Z-axis was measured. The test setup is shown in Figure 3, whereas the interferometer optic was placed on the Z-axis guidance and the mirror on the Z-slide. As a result, it was shown, that the real travel distance exceeds the theoretical one. The surplus travel increases with the number of sine waves and the travel distance (see Figure 4).

Travel path of the Z-axis

Increasing the number of base points per revolution up to $N_s = 7200$ which equals an angular distance of $W_s = 0.05^\circ$, results in a number of revolutions of mere $n = 8 \text{ min}^{-1}$. That means that the machining time is significantly increasing depending on the chosen base point distance. Usually, the feedform has to be described with a significant number of base points which results in realistic angular distances between $W_s = 0.5^\circ$ and $W_s = 1^\circ$ and therefore in numbers of revolution of $n = 83 \text{ min}^{-1}$ and respectively $n = 166 \text{ min}^{-1}$ which means a difference in machining time of the factor 2.

FIGURE 2: Number of revolutions depending on the angular base point distance.



ACKNOWLEDGEMENTS
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SUMMARY AND OUTLOOK
In the paper two effects of slow slide servo were briefly discussed: Firstly, the number of revolutions depending on the base point angular distance, and secondly, the positioning accuracy of the Z-axis while performing sine wave typed travel paths. The presented and ongoing research project aims to identify influences on the workpiece accuracy by knowing the restrictions and limitations of diamond turning with slow slide servo. Next steps will be to identify the influence of the Z-axis travel on the machining time and to engage compensation on travel path deviations by adapted NC-program generation.

FIGURE 4: Deviation of Z-axis travel during slow slide servo.

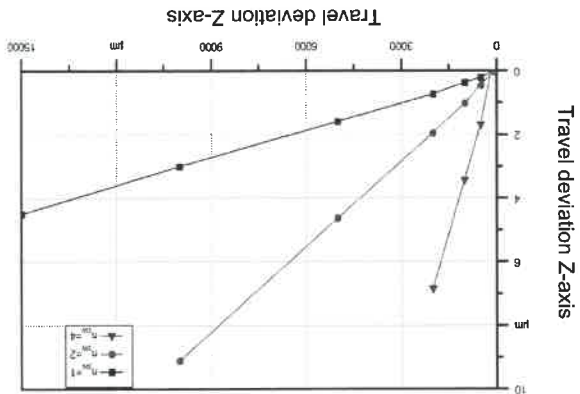
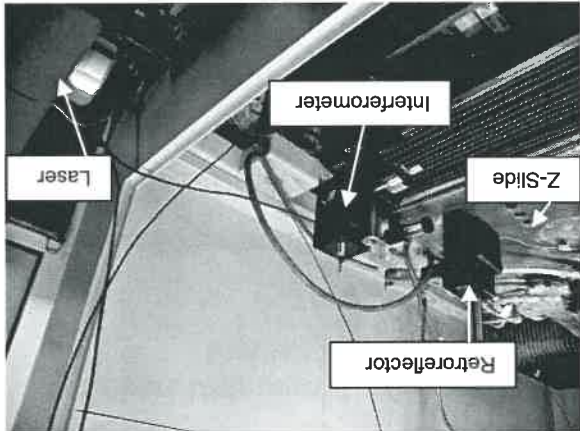


FIGURE 3: Test setup with laser interferometer.



The correlation seems to be linear which leads to the assumption that compensation is possible.

MODELING OF MINIMUM CHIP THICKNESS IN MICROMACHINING OF HARD TOOL STEELS WITH THE MODIFIED SLIP LINE MODEL

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INTRODUCTION

With the ever increasing applications of miniaturized components with high accuracy in a variety of fields, the micromachining process has gained significant importance over the last decade. Researchers have reached the consensus that high aspect ratio and complex geometry 3D structures (made of a variety of materials) are very important. At present, a limited number of technologies retain the capability of manufacturing micro-scale complex 3D shapes with high aspect ratios. In particular, etching-based processes are mainly used to machine silicon and silicon-like materials for MEMS related applications. However, these processes are basically planar 2.5D processes. On the other hand, micro-milling, which is essentially a scaling version of conventional milling processes, represents the most powerful process in terms of versatility and obtainable shapes and features. As the size of the tool and material removal unit are downscaled, there are several critical issues associated with micro-machining and machine tools, which are mainly from the increased requirement of positioning accuracy and rotational speed. The whole volume of material removal is small in comparison to the conventional machining process, while higher tool deflections, high tool wear and high risk of tool breakage are often observed, especially for hard work materials. In die/mold industries, more and more hard materials with high strength are recently used to increase the life of molds, which makes the micromachining process even more challenging. Simultaneously, in order to successfully remove the product, micro-injection molds should have a surface finish RZ equal or less than 1.0 μm . To meet the ongoing performance requirements, the extensive investigation and understanding of the micro-milling process is essential in order to control and improve process capability.

When the chip thickness is comparable in size to the tool edge radius, the bluntness associated with the tool edge creates a 'ploughing' action, which is different from shearing, which is the primary mode of chip formation in micro-machining. Ploughing is considered to be a plastic deformation process where there is displacement of asperities between the two interlocking surfaces. Researchers argued that the total force or specific cutting energy had contributions both from the shearing action and the ploughing action. The specific cutting energy increases with a decrease in the undeformed chip thickness; this is commonly referred to as the "size effect". There are many theories explaining the reasons for the existence of the size effect. Among them are the effect of tool edge radius that causes ploughing, material strengthening effects, subsurface plastic deformation and material separation [1]. The extent of ploughing/rubbing and the nature of the micro-deformation contribute significantly to increased cutting forces [2], burr formation [3] and increased surface roughness [4, 5].

Previous investigators pointed out that the minimum chip thickness effect was the major limiting factor on achievable accuracy because the generated surface roughness is primarily attributed to the ploughing process when the uncut chip thickness is less than the minimum chip thickness [6]. From a physics point of view, both scratching and micromachining generate lateral relative motion of the workpiece through the applied pressure. Essentially both processes are similar to each other. Extensive experimental evidence has shown that the condition for transition from plastic deformation to micro-cutting for a wide range of materials can be estimated by Kragelsky & Dobychin's model [7]. Theoretical and empirical computations by Basuray et al [8] showed that the values of a neutral point angle, which is

Kragelsky [7] was first to consider the criterion for the transition from plastic deformation to microcutting based on indentation test. Kragelsky concluded that microcutting happens when the particles of deforming materials start to adhere to the indenter surface during

and changes to micromachining. zone is formed, i.e., plastic deformation ceases contact surface of the micro-asperities, a dig-in shear resistance of the molecular bond at the body. At a specific value of penetration and surfaces of individual asperities of the harder plastically deforming material flows around the in sliding under plastic contact conditions the

Micromachining

Transition from Plastic Deformation to

hard and brittle materials. minimum chip thickness for micromachining of necessary to develop a method to predict the and workpiece microstructure. Therefore, it is the minimum chip thickness, cutting edge radius affected not only by feed per tooth, but also by process, the quality of machined surface is thickness. Different from a conventional milling machining for its control of minimum chip edge radius condition is critical in micro-rake angle becomes negative. Thus, the cutting ratio is equals to or less than 1.0, the effective chip thickness to cutting edge radius. When the energy is affected by the ratio of undeformed the effective rake angle and specific cutting From previous studies, researchers found that

MODELING OF MINIMUM CHIP THICKNESS

modified slip line theory. based on Kragelsky and Druyanov's model and chip thickness in micro-milling of hard materials method is proposed to determine the minimum desired material removal. In this study, a new radius is essential in micromachining to achieve minimum chip thickness to the cutting edge although the determination of the ratio of brittle materials such as tungsten carbide date has been done on the micro-milling of practical experiments. Moreover, limited work to constitutive material property model from very long simulation time and they also required approaches such as finite element method need uncertainties. On the other hand, numerical accuracy is strongly affected by experimental thickness is tedious and expensive and the method for estimation of minimum chip (angle) are 28~36.7°. However, the experimental equivalent to 90°-α_{cr}, (α_{cr} is the equivalent rake

When the chip thickness is comparable in size to the tool edge radius, the bluntness associated with the tool edge created a 'ploughing' action which is different from shearing, which is the primary mode of chip formation in micro-machining, as shown in Fig. 1. Ploughing is considered to be a plastic deformation process where there is displacement of asperities between the two interlocking surfaces. It is assumed that the rounded tool penetrates an extensive specimen. It has been shown that in such cases, there is negligible upward flow of material and that the displaced material is accommodated by the change in volume that accompanies elastic deformation. In micro-machining, when continuous chips are formed, a

based on Slip Line Theory

Calculation of Minimum Chip Thickness

where h is the limiting depth of penetration of an indenter in scratching, which is equivalent to the minimum chip thickness t_{min} in micromachining, r is the radius of indenter equivalent to the rounded cutting edge radius r , γ is a correction coefficient for the true value of the yield stress in micromachining, σ_f is the effective flow stress of the bulk material and τ_a is the shear strength of the adhesive junction at chip/tool interface. Therefore, to determine the normalized minimum chip thickness λ_m , which is defined as the ratio of the minimum chip thickness to tool edge radius, the effective flow stress of work material and the value of the shear strength of the adhesive junction at chip/tool interface need to be estimated.

$$\frac{r}{h} = \frac{1}{2} \left(1 - \frac{\gamma \sigma_f}{\tau_a} \right) \quad (1)$$

deformation [7]. The flows ceases when than criterion obtained on the basis of planar the conditions of three dimensional deformation thus established gives a better approximation to plane of symmetry of the indenter. The criterion particles of deforming material first occurs in the zone in front of the indenter. Adherence of the the distribution of velocities in the deformation sufficient accuracy, and reduces to determining the asperities on the surface of a body with tapered indenter which reflects the real profile of $r_{max} = k = 1/2(\sigma_1 - \sigma_2)$. The solution is given for a tangential stresses attain the value plastic deformation happens when the maximum According to this condition, the transition to the medium obeys the Tresca flow condition. Druyanov considered both the planar and three-penetration of an indenter into the work material.

tungsten carbide and experiments will also be carried out to verify its effectiveness.

EXPERIMENTAL VERIFICATION

In this study, cemented tungsten carbide (WC) has been tested, whose mechanical properties are elastic modulus of 570.0 GPa, fracture toughness of 8.3 MPa m^{1/2}, and hardness of 17.29 GPa. It is very difficult to machine this brittle material with conventional cutting processes. Two flute PCD cutters as shown in Fig. 2 have been used. During experiments, the spindle rotation speed is kept at a constant value of 120,000 rpm.

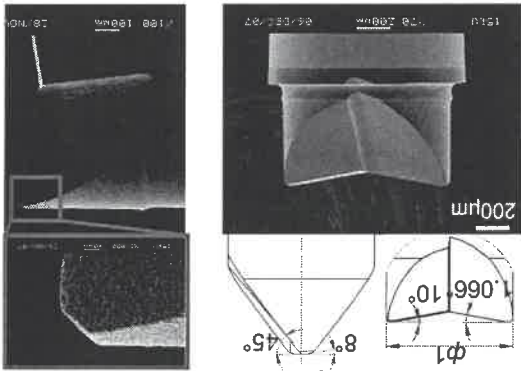


Figure 2. Two flute cutter used in this study.

In Basuray et al [8], the value of a neutral point angle θ is approximately 30° , and the cutting edge radius of a PCD cutter as shown in Fig. 2 is approximately $4.0 \mu\text{m}$. Based on these values and Eq. (4), it can be estimated that the minimum chip thickness is approximately 45.6 nm , when γ is 1.0 .

To verify the effects of minimum chip thickness on the machined surface quality, experiments are carried out. First, one test cut was done with an axial depth of 0.3 μm and feed rate of 0.2 mm/rev-tooth . The machined surface profile is shown in Fig. 3. Theoretically, chips are formed after each revolution since the axial depth of cut is larger than the calculated minimum chip thickness. However, based on the profiles shown in Fig. 3, after approximately 32 μm , the periodic peaks, which mainly affect machined surface quality, are observed. After one cutting pass, the flank surface of the PCD cutter is observed as shown in Fig. 4. One obvious wear land is found on the flank, the reason for this is that the quality of the EDMed cutting edge is not smooth, thus after couple of cutting passes, the cutting edge wears quickly followed by flank wear. Once flank wear appears, a larger area is involved in machining. It acts like a tool with a cutting edge together with a small triangular

small stable built-up edge is formed as shown by the red region in Fig. 1. This built-up edge acts like a cutting tool edge to remove work material. Therefore, the rounded edge tool can be approximated by an equivalent tool with a sharp cutting edge angle and a finite wear land. After this estimation, the effective rake angle varies from 0° to 90° , which depends on the depth of cut. For the given depth of cut of t as shown in Fig. 1, the effective rake angle of the rounded edge tool can be expressed as the angle between the vertical line OA and the tangent BC . The finite wear land AC is obtained from tangents at A and B on the rounded edge. Between ACB and the rounded edge is the stable built-up or dead metal zone. The cutting energy associated with pressure acting on this small stable built-up consists of two parts: one due to the chip sliding over the rake face and the other due to rubbing along the flank face.

For the slip line field for the stagnant friction condition on the rake face as given in Fig. 1, the pressure p action on the face BC can be obtained by applying Hencky's equation:

$$d^2 = -2\phi + d \quad (2)$$

$$(2) \quad z = \phi z + d$$

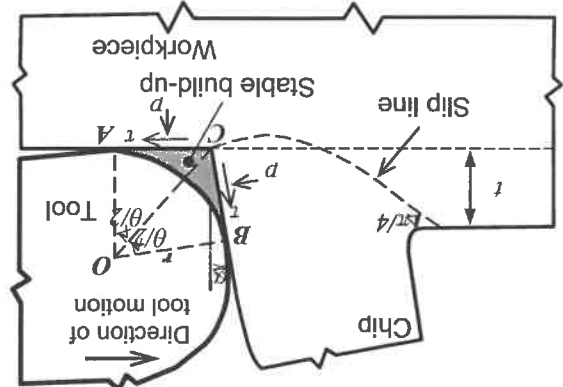


Figure 1. Slip line field for the sticking friction condition on the effective rake face.

Thus the ratio between pressure and shear stress applied on the face BC can be estimated:

$$\frac{\tau}{-p} = \frac{2\phi + 1}{1} \quad (3)$$

$$(3) \quad \frac{1+\phi z}{1} = \frac{z}{d-}$$

Based on Eq. (3), the ratio between the effective flow stress of work material and the value of shear stress at the chip/tool interface can be estimated; thus the ratio of the minimum chip thickness to tool edge radius is obtained as:

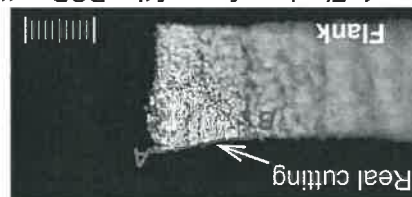
$$\frac{h}{2} = \frac{1}{2} \left(1 - \frac{\phi(2l+1)}{2} \right) \quad (4)$$

This equation will be used to predict the minimum chip thickness for micromachining of

In this study, a modified slip line field model for the stagnant friction condition is developed to estimate the ratio of the effective flow stress of the bulk material to the shear strength of the adhesive junction at chip/tool interface during micromachining of tungsten carbide. First the rounded edge tool can be approximated with an equivalent tool with a straight cutting edge angle and a finite wear land. After this estimation, the effective rake angle can be easily expressed as a function of the depth of cut. Then, by applying Hencky's equation for the simplified geometrical model of cutting tools, the ratio between pressure and shear stress applied on the wear land of the cutting tool can be estimated. Thereafter, this ratio is used for the prediction of minimum chip thickness using Kragelsky and

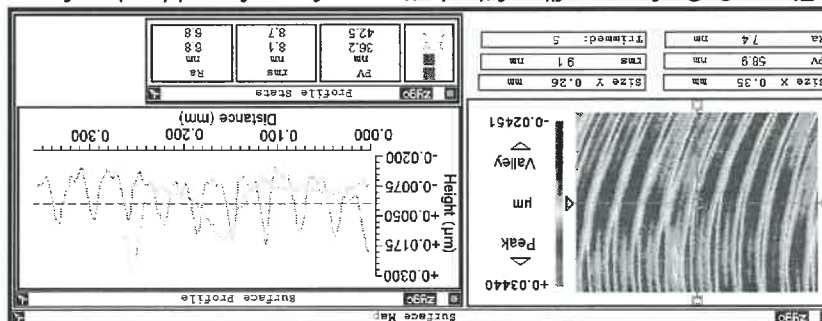
CONCLUSION

Figure 4. Flank surface of the PCD cutter.



area as a grinder. Clearly, from Fig. 3, it can be seen that the marks on the machined surface is mainly generated by one flute. In machining, the cutting edge at Point A removes side and bottom surfaces, then the cutting edge portion between point A and B together with wear land acts like a grinder to press the machined surface and cause the elastic and/or plastic deformation, once the cutting edge at Point B is involved in cutting, chips are formed with the material removal. However, because of the size effect, still some material is left on the machined surface as the sine wave marks on the surface as shown in Fig. 3. That explains that after 32 μm , there are waves with a height around 45nm, which is close to the predicted minimum chip thickness for micromachining of WC workpiece.

Figure 3. Surface profile of the bottom surface of machined surface



Druyanov's prediction model. Finally, experiments were carried out to verify the proposed model for the prediction of minimum chip thickness in micromachining. It is found that the minimum chip thickness for micro-machining of WC with an edge radius of 4 μm is around 45 nm, which is close to the predicted value. Also the minimum chip thickness significantly affects machined surface quality. In order to reduce its effect, it is optimal to set the depth of cut greater than the minimum chip thickness.

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